A. E. Moncaster

THE
MARINE STEAM TURBINE

A PRACTICAL ILLUSTRATED DESCRIPTION

OF THE

PARSONS AND CURTIS MARINE STEAM TURBINES, ETC.

AS PRESENTLY CONSTRUCTED, FITTED, AND RUN

(INCLUDING TURBO-GENERATORS)

A MANUAL OF MARINE STEAM TURBINE PRACTICE

INTENDED FOR THE USE OF

NAVAL AND MERCANTILE MARINE ENGINEER OFFICERS, ETC.

BY

J. W. M. SOTHERN

Principal, Sothern's Marine Engineering College, Glasgow

Member, Institute of Engineers and Shipbuilders in Scotland; Hon. Member, West of Scotland Foremen Engineers' and Draughtsmen's Association; Member, Association of Engineering Teachers

Author of "Verbal Notes and Sketches for Marine Engineers," "Simple Problems in Marine Engineering Design," "Marine Engine Indicator Cards," "Elementary Mathematics," Etc. Etc.

800 Pages, Illustrated by over 700 Diagrams, Photographs, and Detail Drawings

FIFTH EDITION

REWRITTEN UP-TO-DATE AND GREATLY ENLARGED

NEW YORK

D. VAN NOSTRAND COMPANY

25 PARK PLACE

1918

PRINTED IN GREAT BRITAIN BY
THE DARIEN PRESS, EDINBURGH

PREFACE TO FIFTH EDITION.

ERRATA

Page 630.

For—

"1. The consumption (coal or water) (or I.H.P.) varies as the speed.^{2}"

Read—

"1. The consumption (coal or water) (or I.H.P.) varies as the speed.^{3}"

Again, for—

"2. The consumption per knot varies as the speed.^{3}"

Read—

"2. The consumption per knot varies as the speed.^{2}"

A large amount of tonnage (held back meantime owing to Government requirements) has been placed in order for geared down cargo and passenger boats, and this type of machinery promises to be greatly in demand after the war for general service, and will in future replace to some considerable extent the reciprocating class of machinery for merchant service vessels.

In this edition of "The Marine Steam Turbine" a number of new sections have been added which bring the work fully up to date: the original sections have also been carefully revised, and in some cases supplemented by new matter and sketches. The new subjects

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PREFACE TO FIFTH EDITION.

SINCE the previous edition was issued in November 1915 considerable progress has been made in the design and construction of marine turbine machinery, and the improvements effected have resulted in increased efficiency and reduced consumption, which latter, at the present date, is now within measurable distance of about 8½ lbs. of water (or, say, about one pound of coal) per S.H.P. per hour.

The outstanding improvements which have brought about this satisfactory result consist in, first, the extended use of superheat; and second, the developement of gearing; and, in the writer's opinion, the most noteworthy advance of recent years is that due to the introduction of the new double reduction gearing which allows of a very high ratio of gear down between turbine and propeller.

This gearing, it should be mentioned, has already been fitted and run in a number of American vessels, and the results obtained have given every satisfaction.

A large amount of tonnage (held back meantime owing to Government requirements) has been placed in order for geared down cargo and passenger boats, and this type of machinery promises to be greatly in demand after the war for general service, and will in future replace to some considerable extent the reciprocating class of machinery for merchant service vessels.

In this edition of "The Marine Steam Turbine" a number of new sections have been added which bring the work fully up to date: the original sections have also been carefully revised, and in some cases supplemented by new matter and sketches. The new subjects

treated of include:—Impulse Turbines, Turbo-Generators, Double Reduction Gearing, Marine Turbo-electric Drives, Michell Thrust Block, Feed Regulators, Torsion Meters, etc., etc.

The work also contains a large number of new and original sketches and drawings with descriptions, which should be found of value to the student desirous of acquiring reliable information relative to turbine details. A section is also included which treats specially of the practical running of marine turbines, and contains numerous examples of data from actual practice, which it is hoped may be found helpful to engineer officers placed in charge of turbine machinery for the first time.

The thanks of the author are due to Messrs The British Thomson-Houston Co. Ltd., for descriptions and illustrations of turbo-electric machinery, also for information and sketches, etc., relating to Curtis impulse turbines; to Messrs The British-Westinghouse Co. Ltd., for descriptions and illustrations of turbo-electric machinery, and information relating to Rateau turbines; to R. S. Portham, Esq., of Messrs The British Ljungström Marine Turbine Co. Ltd., for information relating to Ljungström turbines; to Messrs The Wall-send Slipway and Engineering Co. Ltd., for illustrations and descriptions of Parsons' turbine machinery; to H. T. Newbigin, Esq., for descriptions and illustrations of the Michell thrust block; to the President and Council of the Liverpool Engineering Society for permission to reprint and reproduce illustrations from the paper of Mr J. Hamilton Gibson, entitled "The Michell Thrust Block and Journal Bearing;" to Messrs Siemens Brothers for descriptions and illustrations of the Hopkinson-Thring torsion meter; to Messrs A. C. Mumford, Ltd., for illustrations and descriptions of their patent feed water regulator; to the editor of the *Mechanical World* for permission to reproduce illustrations of turbine machinery; to the editors of *Engineering* for permission to reproduce and reprint descriptions of turbine machinery. The author's thanks are also specially due to the President and Council of the Institution of Engineers and Shipbuilders in Scotland for kind permission to reprint from the

following papers which have been read previously before the members :—

"The Presidential Address of Alexander Cleghorn, Esq." (delivered October 1917).

"The Ljungström Turbine and its Application to Marine Propulsion." By Roland S. Portham, Esq.

"The Design and Progress of Floating Frame Reduction Gear." By John H. Macalpine, Esq.

"Recent Developments in Air Pump Design." By E. Jones, Esq.

"Some Alternative Types of Propelling Machinery for a 19½-Knot Steamer." By James Dornan, Esq.

The author has also to thank the President and Council of the Institution of Electrical Engineers for kind permission to reproduce illustrations and descriptions of turbo-electric machinery from the paper by K. Baumann, and to Messrs Brady & Martin Ltd., for illustrations and description of their patent kenotometer.

Finally, and as on previous occasions, the author has again to thank many former students for assisting in the compilation of the work by supplying useful data and notes from actual practice and experience.

J. W. M. SOTHERN.

COLLEGE OF MARINE ENGINEERING,
GLASGOW, *Au ust* 1918.

PREFACE TO FOURTH EDITION.

SINCE the issue of the previous edition of this work a number of improvements have developed in relation to marine steam turbine design, the most outstanding being, perhaps, the introduction of gearing between the propeller driving shaft and the turbine to obtain the maximum efficiency of both turbine and propeller independently. Recently, the combined impulse and reaction turbine has also come extensively into marine use, and has given very satisfactory results, particularly in the case of high speed naval vessels.

In Government work a large percentage of the vessels in commission and under construction are fitted with Brown-Curtis type impulse turbines, this design offering special advantages for the conditions obtaining in naval practice. The Parson's geared-down turbine appears to be gradually displacing both the combined reciprocating and turbine arrangement and the direct drive turbine, as the gearing has given every satisfaction under severe service conditions in steamers of large displacement and power: the innovation may therefore be looked upon as having come to stay. Regarding the present edition of the "Marine Steam Turbine," it has to be stated that the new matter and illustrations introduced (Sections V.-XV.) refers chiefly to impulse turbines and geared-down turbines; in addition to this, a short section has been given over to the discussion of entropy, a subject not of such common knowledge among marine engineers as might be desired.

The Author has taken special care to make as clear as possible the explanations and descriptions of the velocity of steam through the blades, etc., and has, in fact, laid himself open to the charge of over-elaboration and repetition; but from experience this close atten-

tion to detail has been found necessary for the clear understanding of the subject to the reader devoid of turbine engineering knowledge.

The Author has to thank numerous former students and friends who have most kindly supplied data from actual practice, some of which, it may be pointed out, was taken specially for the purposes of the book. The Author's thanks are also due to the Council of the Institution of Engineering and Shipbuilders in Scotland for permission to reproduce illustrations of the "Otaki"; to the Council of the North-East Coast Institution of Engineers and Shipbuilders for illustrations of the "Cairnross" turbines; to the Council of the Institution of Naval Architects for illustrations of the "Vespasian" gear; to the Editors and Proprietors of *Engineering* for permission to reproduce numerous illustrations; and to Messrs G. & J. Weir, Ltd., for illustrations of their auxiliary gear and condensers.

The Author has also to thank his assistants, W. A. C. Fry, Esq., and R. M.. Sothern, Esq., M.I.E.S., for their able work in revising and checking the calculations, and for the drawing out of a number of the line illustrations and diagrams.

In conclusion, it may be mentioned that, as in former editions, the data given in the present volume can be relied upon as being accurate and trustworthy.

59 Bridge Street, Glasgow,
1915.

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THE MARINE STEAM TURBINE.

SECTION I.

DEFINITIONS AND GENERAL PRINCIPLES.

BEFORE commencing to describe the chief features of the Parsons turbine, it is perhaps necessary to explain clearly the meaning of certain definitions which are closely connected with the theory and practice of this type as with all other types of steam engine.

Foot-Pound.—A foot-pound is the work done in raising a weight of one pound up through a distance of one foot.

Torque.—Torque is the turning movement to which a shaft is subjected when a force is exerted to rotate the shaft against a resistance such as that of the screw propeller in water. In ordinary engines the turning effort or torque is applied by means of the crank, and in turbines by the direct energy of the steam acting on the periphery of the blade circle of the rotor.

By arranging the rotor diameter so that the peripheral velocity of the blades is equal to about half that of the steam, the maximum amount of work in foot-pounds may be extracted from each pound of steam passing through the turbine casing.

Heat.—Heat is merely a form of energy, and as such exists in two states—(1) in that of Potential or stored-up energy, and (2) in that of Kinetic or active energy. When the molecules of a body or

gas are set in rapid motion or vibration, heat is developed and work done. Consequently in the case of a steam engine, either of the reciprocating type or turbine type, the energy which produces rotation of the shaft is obtained by means of the transformation or heat energy into mechanical work.

British Thermal Unit (B.Th.U.).—This is taken as being equal to 778 foot-pounds of work or energy, and signifies that one heat unit, when transformed into mechanical energy, gives out 778 foot-pounds of work.

Saturated Steam.—Steam taken direct from the boilers is known as "saturated steam," as the density, or weight of water per cubic foot, is constant for any given pressure, as also is the temperature and volume. The steam supplied to all marine engines (without superheaters) is therefore of this quality, and calculations as to expansion, work done, and fall of pressure, are usually made on this assumption. The steam supplied to the H.P. turbine of a turbine engine is therefore saturated steam. Sometimes the term "dry saturated steam" is used to distinguish this quality of steam from wet steam, or steam containing water from priming.

"Wet" Steam.—If water is carried off with the steam due to priming taking place in the boilers, the steam contains more water per cubic foot than is natural to the "saturation" pressure, volume, and temperature, and it is then known as "wet steam," or "wet saturated steam."

Superheated Steam.—If saturated steam from the boilers is passed through the tubes of a superheater, the water contained in the steam is evaporated out of it, with the following results:—

1. Rise of temperature.
2. Increase of volume if pressure is kept constant; or,
3. Increase of pressure if volume is kept constant.

The chief advantage of superheated steam lies in the fact that cylinder condensation is practically eliminated, as the steam does not then readily condense when exposed to cooled surfaces: leakage is also reduced.

Another point of importance is that the specific heat of this steam being only .48 (some authorities give .5), one B.T.U. of heat supplied to the steam has the effect of raising its temperature fully two degrees, as $1 \div .48 = 2.08$.

Properties of Saturated Steam

Of from 0.5 lb. to 260 lbs. Absolute Pressure per Square Inch.

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° F.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
0.5	80.2	1105.5	1058.4	.001376	726.608
1	102.1	1112.5	1042.9	.003027	330.360
1.5	115.9	1116.7	1033.2	.004433	225.580
2	126.3	1119.9	1025.8	.005811	172.080
2.5	134.6	1122.5	1019.9	.007169	139.488
3	141.6	1124.6	1015.0	.008511	117.500
3.5	147.7	1126.4	1010.6	.009839	101.632
4	153.1	1128.1	1006.8	.01116	89.632
4.5	157.9	1129.6	1003.4	.01246	80.231
5	162.3	1130.9	1000.3	.01370	72.991
5.5	166.4	1132.1	997.4	.01505	66.428
6	170.2	1133.3	994.7	.01634	61.201
6.5	173.6	1134.3	992.3	.01762	56.761
7	176.9	1135.3	990.0	.01889	52.936
7.5	180.0	1136.3	987.8	.02016	49.610
8	182.9	1137.2	985.7	.02142	46.686
8.5	185.7	1138.0	983.8	.02268	44.097
9	188.3	1138.8	981.9	.02394	41.777
9.5	190.8	1139.5	980.1	.02547	39.261
10	193.3	1140.3	978.4	.02642	37.845
10.5	195.6	1141.0	976.7	.02767	36.145
11	197.8	1141.7	975.2	.02890	34.599
11.5	200.1	1142.4	973.6	.03026	33.045
12	202.0	1143.0	972.2	.03137	31.879
12.5	204.0	1143.6	970.8	.03260	30.678
13	205.9	1144.2	969.4	.03382	29.573
13.5	207.8	1144.8	968.1	.03504	28.536
14	209.6	1145.3	966.8	.03627	27.573
14.7	212.0	1146.1	965.2	.03797	26.360
15	213.1	1146.4	964.3	.03870	25.843
16	216.3	1147.4	962.1	.04112	24.320
17	219.6	1148.3	959.8	.04253	23.513
18	222.4	1149.2	957.7	.04594	21.766
19	225.3	1150.1	955.7	.04834	20.687
20	228.0	1150.9	953.8	.05074	19.710
21	230.6	1151.7	951.9	.05311	18.828
22	233.1	1152.5	950.2	.05549	18.022
23	235.5	1153.2	948.5	.05786	17.282
24	237.8	2153.9	946.9	.06023	16.603
25	240.1	1154.6	945.3	.06259	15.977
26	242.3	1155.3	943.7	.06495	15.401

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
27	244.4	1155.8	942.2	.06728	14.863
28	246.4	1156.4	940.8	.05971	14.345
29	248.4	1157.1	939.4	.07196	13.896
30	250.4	1157.8	937.9	.07430	13.459
31	252.2	1158.4	936.7	.07663	13.050
32	254.1	1158.9	935.3	.07894	12.666
33	255.9	1159.5	934.0	.08128	12.300
34	257.6	1160.0	932.8	.08358	11.964
35	259.3	1160.5	931.6	.08590	11.640
36	260.9	1161.0	930.5	.08821	11.337
37	262.6	1161.5	929.3	.09050	11.050
38	264.2	1162.0	928.2	.09282	10.773
39	265.8	1162.5	927.1	.09510	10.515
40	267.3	1162.9	926.0	.09740	10.267
41	268.7	1163.4	924.9	.09946	10.054
42	270.2	1163.8	923.9	.1020	9.806
43	271.6	1164.2	922.9	.1042	9.592
44	273.0	1164.6	921.9	.1065	9.386
45	274.4	1165.1	920.9	.1088	9.191
46	275.8	1165.5	919.9	.1111	9.003
47	277.1	1165.9	919.0	.1134	8.821
48	278.4	1166.3	918.1	.1156	8.650
49	279.7	1166.7	917.2	.1179	8.482
50	281.0	1167.1	916.3	.1202	8.322
51	282.3	1167.5	915.4	.1224	8.170
52	283.5	1167.9	914.5	.1247	8.021
53	284.7	1168.3	913.6	.1269	7.880
54	285.9	1168.6	912.8	.1292	7.741
55	287.1	1169.0	912.0	.1314	7.610
56	288.2	1169.3	911.2	.1337	7.482
57	289.3	1169.7	910.4	.1357	7.370
58	290.4	1170.0	909.6	.1382	7.238
59	291.6	1170.4	908.8	.1404	7.123
60	292.7	1170.7	908.0	.1426	7.011
61	293.8	1171.1	907.2	.1449	6.902
62	294.8	1171.4	906.4	.1471	6.798
63	295.9	1171.7	905.6	.1493	6.696
64	296.9	1172.0	904.9	.1516	6.596
65	298.0	1172.3	904.2	.1538	6.502
66	299.0	1172.6	903.5	.1560	6.410
67	300.0	1172.9	902.8	.1583	6.318
68	300.9	1173.2	902.1	.1604	6.233
69	301.9	1173.5	901.4	.1627	6.147

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density of Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	L s.	Cubic Feet.
70	302.9	1173.8	900.8	.1650	6.059
71	303.9	1174.1	900.3	.1671	5.984
72	304.8	1174.3	899.6	.1693	5.905
73	305.7	1174.6	898.9	.1716	5.829
74	306.6	1174.9	898.2	.1738	5.764
75	307.5	1175.2	897.5	.1760	5.683
76	308.4	1175.4	896.8	.1782	5.610
77	309.3	1175.7	896.1	.1803	5.544
78	310.2	1176.0	895.5	.1826	5.476
79	311.1	1176.3	894.9	.1848	5.411
80	312.0	1176.5	894.3	.1870	5.348
81	312.8	1176.8	893.7	.1892	5.286
82	313.6	1177.1	893.1	.1912	5.230
83	314.5	1177.4	892.5	.1936	5.167
84	315.3	1177.6	892.0	.1957	5.109
85	316.1	1177.9	891.4	.1980	5.052
86	316.9	1178.1	890.8	.2001	4.996
87	317.8	1178.4	890.2	.2023	4.942
88	318.6	1178.6	889.6	.2046	4.889
89	319.4	1178.9	889.0	.2067	4.837
90	320.2	1179.1	888.5	.2088	4.790
91	321.0	1179.3	887.9	.2111	4.737
92	321.7	1179.5	887.3	.2133	4.688
93	322.5	1179.8	886.8	.2154	4.642
94	323.3	1180.0	886.3	.2176	4.595
95	324.1	1180.3	885.8	.2198	4.549
96	324.8	1180.5	885.2	.2220	4.505
97	325.6	1180.8	884.6	.2241	4.462
98	326.3	1181.0	884.1	.2263	4.419
99	327.1	1181.2	883.6	.2286	4.375
100	327.9	1181.4	883.1	.2307	4.335
101	328.5	1181.6	882.6	.2329	4.305
102	329.1	1181.8	882.1	.2350	4.256
103	329.9	1182.0	881.6	.2372	4.216
104	330.6	1182.2	881.1	.2393	4.178
105	331.3	1182.4	880.7	.2415	4.140
106	331.9	1182.6	880.2	.2437	4.104
107	332.6	1182.8	879.7	.2458	4.068
108	333.3	1183.0	879.2	.2480	4.033
109	334.0	1183.3	878.7	.2502	3.998

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supposed at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
110	334.6	1183.5	878.3	.2523	3.963
111	335.3	1183.7	877.8	.2545	3.930
112	336.0	1183.9	877.3	.2566	3.897
113	336.7	1184.1	876.8	.2588	3.865
114	337.4	1184.3	876.3	.2610	3.832
115	338.0	1184.5	875.9	.2631	3.801
116	[illegible]	1184.7	875.5	.2653	3.770
117	339.3	1184.9	875.0	.2674	3.740
118	339.9	1185.1	874.5	.2696	3.710
119	340.5	1185.3	874.1	.2717	3.681
120	341.1	1185.4	873.7	.2738	3.652
121	341.8	1185.6	873.2	.2760	3.623
122	342.4	1185.8	872.8	.2781	3.595
123	343.0	1186.0	872.3	.2803	3.567
124	[illegible]	1186.2	[illegible]	.2824	3.541
125	344.2	1186.4	871.5	.2845	3.514
126	344.8	[illegible]	871.1	.2867	3.488
127	345.4	[illegible]	870.7	[illegible]	3.462
128	[illegible]	[illegible]	[illegible]	.2910	3.436
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3.411
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3.388
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Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
152	359.5	1191.0	860.7	.3421	2.923
153	360.0	1191.2	860.4	.3442	2.905
154	360.5	1191.4	860.0	.3463	2.887
155	361.1	1191.5	859.6	.3484	2.870
156	361.6	1191.7	859.2	.3505	2.853
157	362.1	1191.8	858.9	.3527	2.836
158	362.6	1192.0	858.5	.3548	2.818
159	363.1	1192.1	858.1	.3569	2.802
160	363.6	1192.3	857.8	.3590	2.785
165	366.0	1192.9	856.2	.3696	2.706
170	368.2	1193.7	854.5	.3801	2.631
175	370.8	1194.4	852.9	.3905	2.559
180	372.9	1195.1	851.3	.4011	2.493
185	375.3	1195.8	849.6	.4115	2.430
190	377.5	1196.5	848.0	.4220	2.370
195	379.7	1197.2	846.5	.4324	2.313
200	381.7	1197.8	845.0	.4419	2.263
210	286.0	1199.1	841.9	.463	2.157
220	389.9	1200.3	839.2	.484	2.065

Adiabatic Expansion.—If steam expands in a cylinder or turbine casing, and neither receives heat from any external source nor gives out any heat externally, then the expansion is said to be "adiabatic," and all work done in the cylinder or turbine is obtained at the expense of the internal heat of the steam, which in falling in pressure and temperature conforms to this condition, and part of which condenses. In the cylinders of a marine engine of the reciprocating type, the expansion is approximately hyperbolic or isothermal, and in a turbine the expansion is approximately "adiabatic."

Hyperbolic or Isothermal Expansion.—This is founded on the well-known law of Boyle and Marriot that the pressure of a gas varies inversely as the volume; or, as it is expressed—

Rule, $P_1 \times v_1 = p_2 \times V_2 =$ Constant.

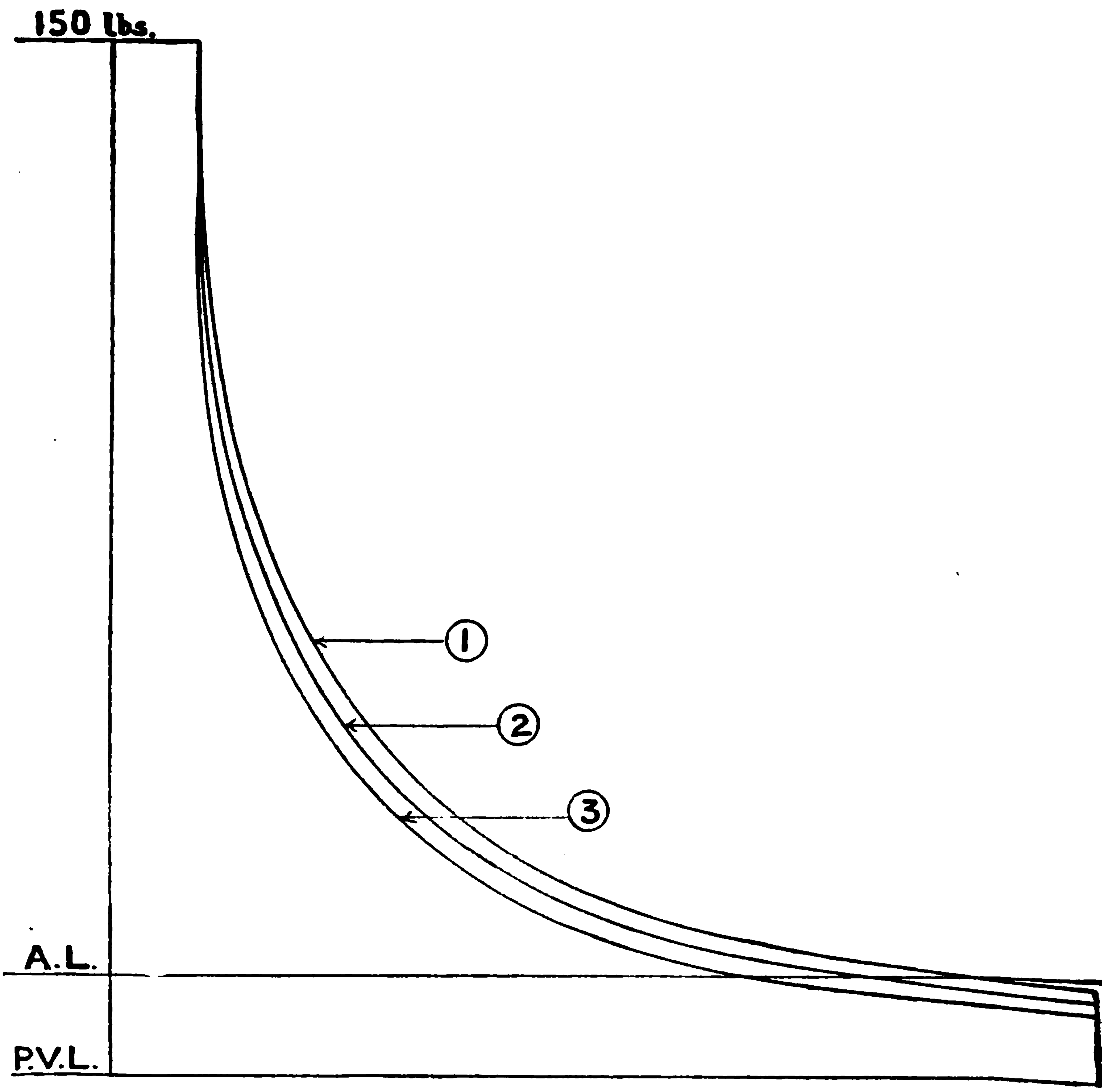
Where P_1 = Initial pressure. p_2 = Final pressure.
v_1 = ,, volume. V_2 = ,, volume.

$$\text{Therefore, } \frac{P_1 \times v_1}{p_2} = V_2;$$

$$\text{and, } \frac{P_1 \times v_1}{v_2} = p_2;$$

$$\text{or, } \frac{p_2 \times V_2}{P_1} = v_1;$$

$$\text{and, } \frac{P_2 \times V_2}{v_1} = P_1$$



Expansion Curves of Steam.

(1) Isothermal or Hyperbolic Curve, $P \times V = \text{Constant}$ (Perfect Gas).
(2) Saturation Curve, $P \times V^{\frac{17}{16}} = \text{Constant}$ (Reciprocating Engine, approximately).
(3) Adiabatic Curve, $P \times V^{\frac{10}{9}} = \text{Constant}$ (Turbine Engine, approximately).

As explained before, ordinary steam, being an imperfect gas, does not exactly follow out the above law, but deviates in the direction of the "saturated steam" curve, as shown in the diagrams, in the case of reciprocating engine cylinders.

Dryness Fraction (or Factor).—In considering the actual work done by steam between the rows of blades of a turbine, it is important that the dryness fraction be taken into account, as the result greatly depends on this quantity. After work is done by adiabatic expansion, the steam contains a certain amount of water, which proportionally reduces the internal heat still left in the steam. The dryness fraction is the ratio between the weight of dry steam per pound and the weight of the dry steam and water added together;

$$\text{Or, } \frac{\text{Weight of dry steam}}{\text{Weight of dry steam} + \text{weight of water}} = \text{Dryness Fraction.}$$

Suppose the water to be 25 per cent. of each pound weight of mixture,

$$\text{Then, } \frac{100-25}{100} = \frac{75}{100} = \frac{15}{20} = \frac{3}{4} = \text{Dryness Fraction (or Factor).}$$

So that after expansion and work done by the steam the actual units or foot-pounds of energy left are equal to the internal heat units multiplied by the fraction $\frac{3}{4}$.

Total Heat of Steam.—By the total heat of saturated, or boiler steam, is meant the number of heat units required to produce one pound of steam from a temperature of 32° Fahr. to any given temperature and pressure. The total heat includes the latent heat of steam formation and the sensible or thermometer heat.

RULE. $1083 + .3 \times T^{\circ} =$ Total Heat (above 32° Fahr.).

Where $T^{\circ} =$ Temperature of the steam (Fahr.).

Internal Heat of Steam.—By this is meant the heat or energy required to change one pound of water into steam at any given pressure.

External Heat of Steam.—By this is meant the heat required to produce increase of volume (water to steam) against an external resistance or pressure.

Latent Heat of Steam.—The sum of the Internal heat and External heat is equal to the latent heat.

The Latent Heat can be calculated as follows:—

RULE. $1114 - .7 \times T^{\circ} =$ Latent Heat.

Where $T^{\circ} =$ Temperature of the steam (Fahr.).

EXAMPLE.—Calculate the Total Heat, Latent Heat, and Sensible Heat of 1 lb. of steam at 160 lbs. pressure by gauge.

$160 + 15 = 175$ lbs. absolute pressure and 371° Temperature (from Table, page 3),
Then, $1083 + .3 \times 371 = 1194.3$ Total Heat,
and $1114 - .7 \times 371 = 854.3$ Latent Heat.
Therefore $371° - 32° = 333.9$ Sensible Heat.

NOTE.—The above are all calculated from a temperature of 32° Fahr.

Potential Energy is the energy contained or stored up in steam of a given pressure and temperature, the amount of energy contained increasing with the pressure and the temperature.

Kinetic Energy is the result of setting free the potential or stored-up energy of the steam, which then shows as active energy in the performance of work. In a steam engine the steam acts on the pistons, and by causing motion to take place work is done, and, as a result, the steam falls in pressure and in temperature. In a turbine, the steam at a given pressure and velocity leaves the first row of guide blades, and striking the first row of moving blades gives up part of its kinetic energy, which results in a decrease in pressure and in heat. It then enters the next row of guide and moving blades, where more energy is given up, and a further decrease in pressure and in heat takes place. This is repeated row after row, the steam falling in pressure and in temperature, but, be it noted, increasing in volume. This increase of volume would produce increase of velocity if the blades were not made (1) longer, or (2) spaced farther apart. Both methods, separately and combined, are adopted at different expansion stages of the Parsons turbine, as will be shown later.

The effective kinetic energy of each pound of steam supplied to the turbine is employed in exerting a torsional stress on the shaft, and thus produces rotation. In the boilers the potential energy of the steam is generated, and in the cylinders of an ordinary engine, or in the turbine casing of a turbine engine, the potential energy is liberated and transformed into kinetic energy, and a certain number of foot-pounds of work are done in causing rotation of the shaft. In all steam engines only a very small portion of the total heat of the steam can be changed into foot-pounds of useful work, seldom more than about 14.3 per cent., or $\frac{1}{7}$, as shown by the following:—

Suppose the consumption of coal is 1.5 lb. per I.H.P. per hour—

$$\text{Then, } \frac{60 \times 33000}{1.5 \times 11500 \times 778} = \frac{1}{7}, \text{ or } 14.3 \text{ per cent.}$$

NOTE.—Allowing 11500 units of heat to be given up per pound of coal.

This applies equally to a turbine or reciprocating engine. Suppose, then, that a pound of steam in passing through a turbine gives up, say, 200 units of heat,

Then, $200 \times 778 = 155600$ foot-pounds of energy given out.

This means that 155600 foot-pounds of work have been done in rotating the shaft, neglecting blade leakage, blade friction, and other losses.

If, then, we know that a certain number of pounds of steam are used in a given time, say, for example, 1000 lbs. per minute, then,

$$1000 \times 155600 = 155600000 \text{ foot-pounds of energy developed.}$$

After losses are allowed for, the actual foot-pounds of work done, or kinetic energy expended, is exactly equal to the units of heat effectively applied, multiplied by 778.

Blade Friction.—It is important that the surfaces of the blades should be as smooth as possible, since the friction produced by the flow of steam across the blades somewhat seriously affects the efficiency of the turbine. It will also be obvious that water in the steam will produce a similar result, so that one clear advantage of superheating the steam is the reduction in frictional resistance set up by saturated steam.

Expansion of Steam.—In a modern marine triple-expansion engine with cylinder areas of H.P. to L.P. as 1 is to 7.5, and with a "linked up" cut-off in the H.P. of $\frac{1}{3}$ stroke, the total number of expansions of steam would be 22.5, as $7.5 \times 3 = 22.5$ (by volumes).

In the turbine engine, however, the number of expansions of steam is much more than this, from 125 to 140 expansions being readily obtained. With an H.P. turbine initial pressure of 150 lbs. absolute, and a condenser vacuum of 29 in., or back pressure of say 1 lb., the steam would expand about 150 times by pressures, as $150 \div 1 = 150$. From this it will be seen that more work can be got out of the steam by the greatly increased number of steam expansions, and the importance of obtaining a very high vacuum in the condensers will be obvious.

Condensation.—In all ordinary engine cylinders losses from condensation are more or less serious, the cause being, as is well known the difference of temperature existing between the initial and exhaust pressures, alternately heating up and cooling down the cylinder walls. This results in a certain amount of steam condensing into water without doing work.

In a turbine casing no such variation in temperature exists, as the range of temperature is practically constant throughout the turbine from end to end, the steam entering at one end at a high temperature, and flowing to the other end continuously as it falls gradually in pressure and temperature.

Again, if condensation does take place, the water formed is not so troublesome to get rid of as in an ordinary cylinder, as it simply drains away to the exhaust end of the turbine and so to the air pumps.

The condensation which does occur in the turbines is that due to the adiabatic expansion of the steam during the transformation from potential to kinetic energy, as described elsewhere.

Principle of Turbine.—The steam turbine is a machine designed to convert the kinetic energy of steam into direct rotary motion. The two principal types of turbine are—(1) Impulse Turbines, those arranged with expanding nozzles in which the high velocity of discharge impinges against a series of small buckets secured on the circumference of a large wheel keyed to the driving shaft, the De-Laval turbine being an example of this type; and those (2) Impulse-reaction Turbines, in which the steam passes through a number of rings of fixed blades and of moving blades, expanding as it travels, an example of which is found in the Parsons turbine.

Work by impulse is produced by high velocities, and as the work is done at the expense of the internal heat, water is formed, which thus diminishes the heat left.

The Parsons turbine is generally called a reaction turbine, although the correct term should be "impulse-reaction" turbine, as the steam actually does act first by impulse from the guide to the moving blades and afterwards works by reaction from the moving to the guide blades.

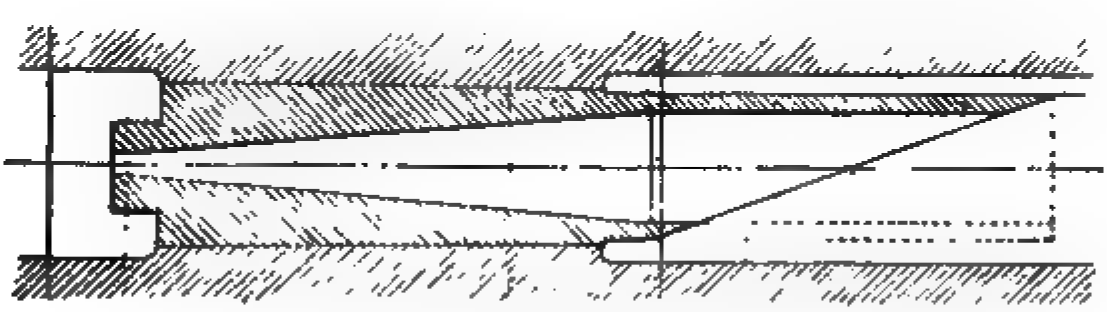
ARRANGEMENT OF NOZZLE AND SHUTTING-OFF VALVE.
De-Laval Turbine.

De-Laval Turbine.—In the specially shaped diverging nozzle of the De-Laval turbine shown in the sketch, the steam expands down to the required exhaust pressure, and the resultant kinetic energy acquired is applied direct to the small buckets or vanes, the steam being in consequence at a very high velocity. To obtain the best efficiency the circumferential velocity of the turbine blades should be equal to about half the velocity of the steam, and this, of course, demands a very high revolution speed. In the De-Laval turbine the speed is often as high as 20,000 revolutions per minute; this can, however, be reduced by suitable gearing to about 2,000 revolutions per minute, but as even this is too high for the shafting of marine engines, the non-adaptability of this turbine for marine purposes will be obvious. The steam is admitted to the nozzles (usually four or six in number) and controlled by regulating hand valves.

It is worthy of notice that in this type of turbine the turbine wheel is rotated by steam at the expanded or lowest pressure, as the actual

View of De-Laval Turbine in Action.

expansion takes place in the nozzle (and not within the vanes or buckets), which is specially designed for that purpose.



Nozzle of De-Laval Turbine.

The De-Laval type of turbine is much in use for the driving of dynamos, and many steamers are supplied with this turbine for the lighting set of the ship.

Parsons Turbine.—In this, the latest and most successful development of marine engineering, steam is admitted direct from the boilers to blades on the shaft drum, thus doing away with the necessity for piston valves or slide valves, cylinders, pistons, piston rods, crossheads, connecting rods, cranks, eccentrics, eccentric rods, and links, &c.

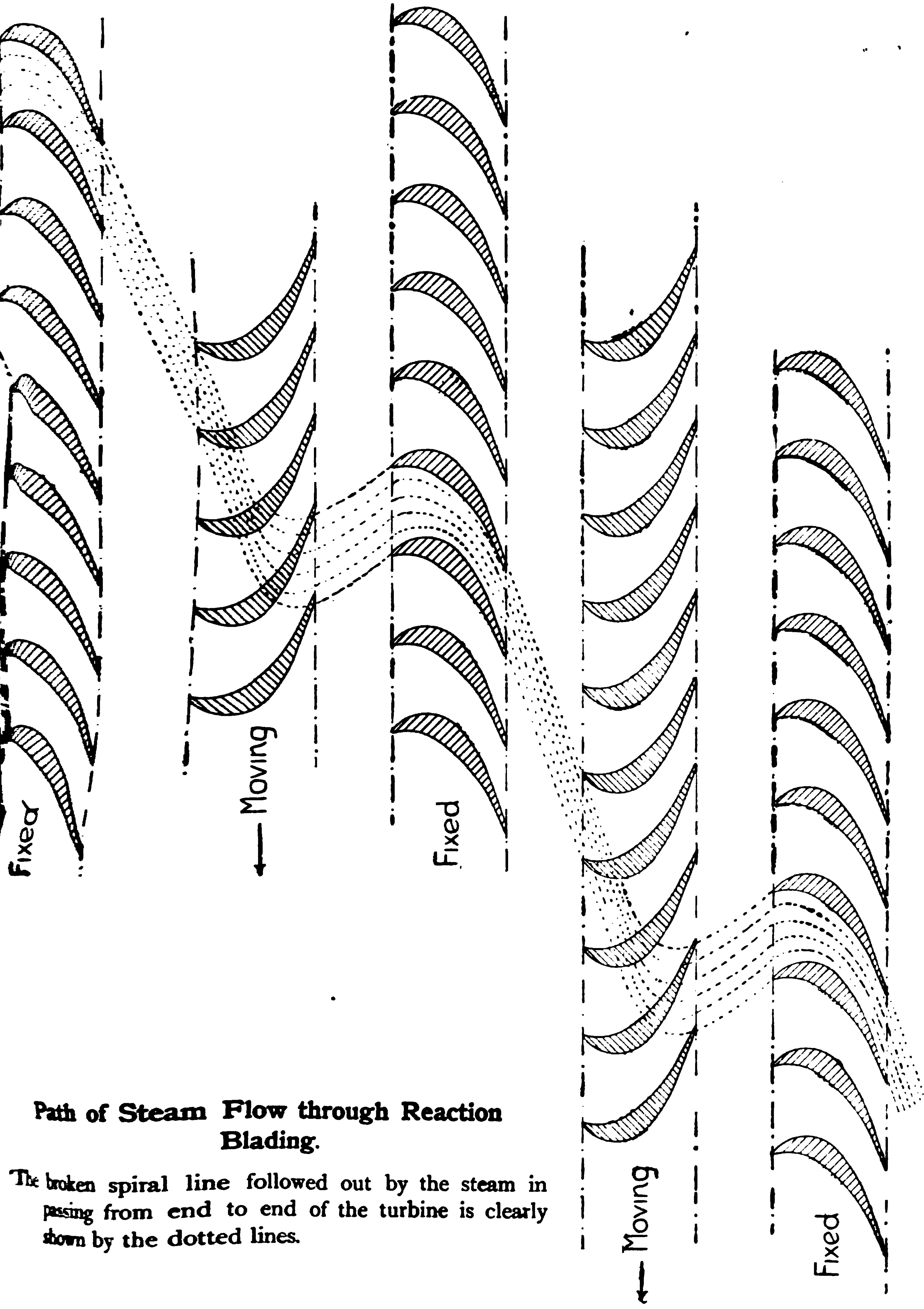
The power to rotate the shaft is therefore applied direct, which in itself constitutes one of the conditions of an ideal engine. The inventor, the Hon. C. A. Parsons, M.A., F.R.S., gives the following brief description of the turbine:—

"The Parsons turbine consists of a cylindrical case with numerous rings of inwardly projecting blades. Within this cylinder, which is of variable internal diameter, is a shaft or spindle, and on this spindle are mounted blades, projecting outwardly, by means of which the shaft is rotated. The former are called fixed or guide blades, and the latter revolving or moving blades. The diameter of the spindle is less than the internal diameter of the cylinder, and thus an annular space is left between the two. This space is occupied by the blades, and it is through these the steam flows. The steam enters the cylinder by means of an annular port at the forward end; it meets a ring of fixed guide blades which deflect it so that it strikes the adjoining ring of moving blades at such an angle that it exerts on them a rotary impulse. When the steam leaves these blades it has naturally been deflected. The second ring of fixed blades is therefore interposed, and these direct the steam on to the second ring of rotating blades. The same thing occurs with succeeding rings of guide and moving blades until the steam escapes at the exhaust passage."

Steam from the boiler is admitted by suitable hand valves to the forward end of the casing surrounding the blades, and after passing through a ring of guide blades fixed to the casing, strikes the first ring of shaft or rotor blades; it next passes through the second ring of fixed blades, then the second ring of rotor blades, and so on, passing alternately ring after ring of guide and rotor blades, and so rotating the shaft, until it finally exhausts at the other end of the turbine casing at a reduced pressure.

Parallel Flow.—Parsons' marine turbine is known as that of the impulse and reaction "parallel flow" type, as the steam enters the guide vanes in lines parallel more or less to the shaft axis, and in this way passes from end to end of the turbine, reacting, expanding, and falling in pressure as it travels.

Action of Steam.—As will be seen from the foregoing, the steam striking the blades imparts a turning movement to the shaft, and after reacting and passing through the series of rings of vanes of the H.P. turbine exhausts simultaneously into the two L.P. turbines, one on either side, and expanding through the longer casing and shaft blades of these turbines, finally exhausts, at a low absolute pressure of from 1½ to 2 lbs., into the condensers, one for each L.P. turbine.



Path of Steam Flow through Reaction Blading.

The broken spiral line followed out by the steam in passing from end to end of the turbine is clearly shown by the dotted lines.

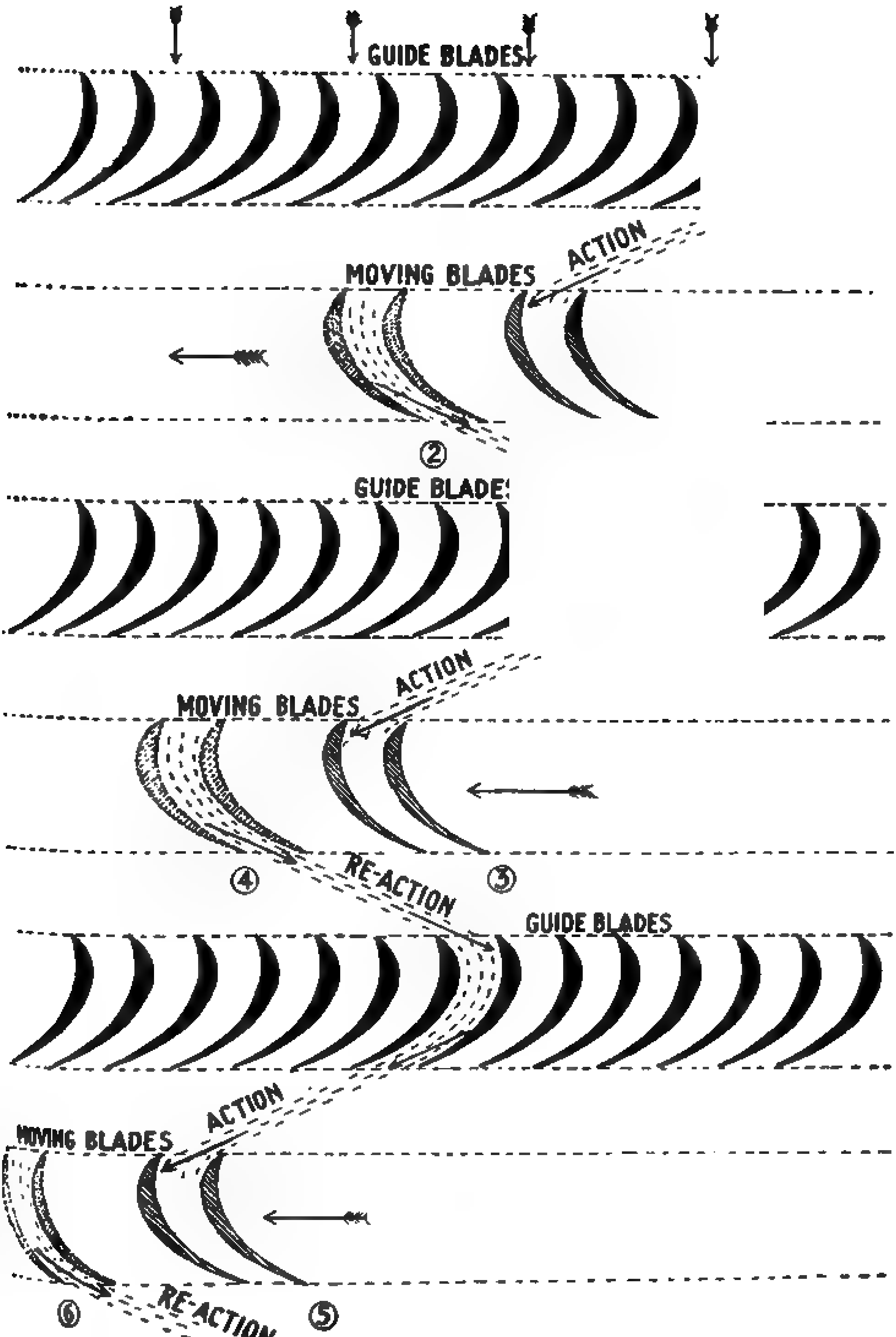


Diagram showing Action and Reaction of Steam on Blades.

B

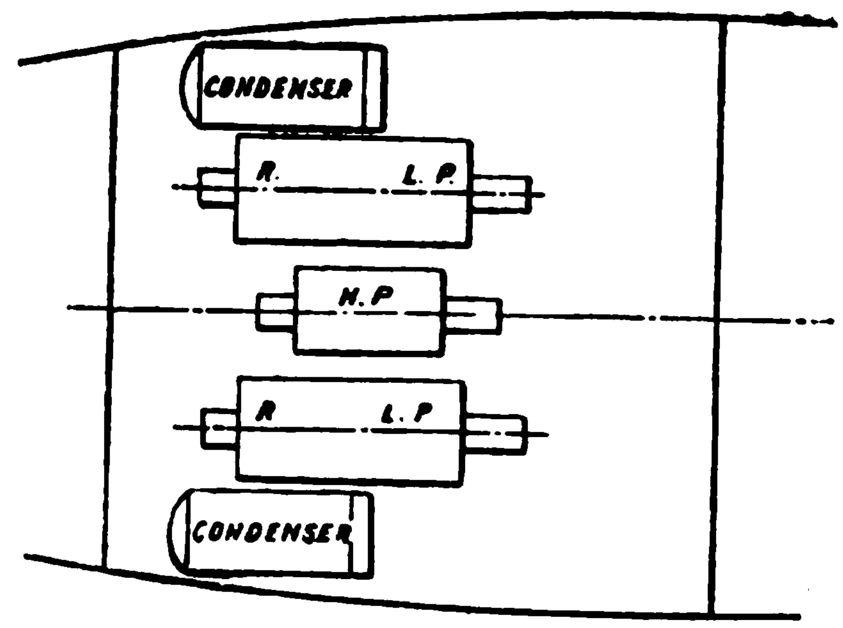
Flow of Steam through Blades.—The diagram on page 15 shows graphically the path followed out by the steam as it passes through each successive ring of fixed and moving blades. Observe that the steam, after passing through the first ring of guide blades, strikes the first ring of rotor or moving blades and by the action set up assists in rotating the shaft; by the time the steam has changed its direction the rotor has moved round a certain distance (from 1 to 2), and the reaction of the steam, due to its somewhat sudden change of direction, still further assists in rotating the shaft. The steam then leaves the rotor blades and enters the next ring of guide blades, where, after again being deflected in its path, it enters the next ring of moving blades, where the action and reaction process is again repeated; leaving the second ring of moving blades at position 4 the steam enters the third ring of guide blades at a point 5 still farther round the circumference, and so on for each of the following rings.

It should be noted that the steam leaving the moving blades is deflected by the blade curvature, and strikes the casing blades, which, if free to revolve, would be acted on by the steam and moved round similarly to the rotor blades, but in the opposite direction; instead of this taking place, however, the casing blades being fixed resist the impact, and the resulting reaction throws back the steam, the velocity of which is thus increased. The pressure is therefore utilised in augmenting the steam speed, hence the statement that "in the guide blades the steam does work on itself to increase its own velocity." The steam thus describes a somewhat zigzag path in passing along the rotor, its direction being not unlike that of a screw thread. Work is done at each ring of blades and heat given up, expansion of the steam taking place in due proportion, so that the velocity of flow increases, and to allow for this the lengths and spacings of the blades must be increased to maintain the same ratio between the blade velocity and the steam velocity, upon which the turbine efficiency depends.

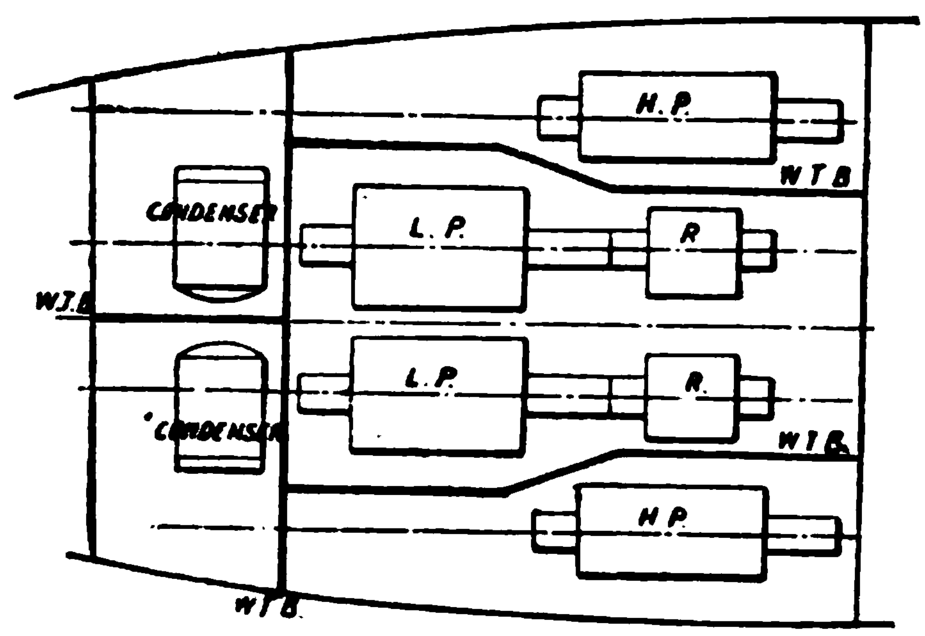
The diagram shows the imaginary path described by a small portion of steam, and the dotted blades show the circumferential advance of the rotor blades at each ring, which produces the thread-like path traced out by the steam.

Turbine Arrangements.—In steamers of ordinary size for either channel or deep-sea service, the standard arrangement consists of five turbines, three for ahead and two for reverse running; three shafts are fitted with one propeller on each, the reverse turbines being placed within the L.P. turbine casings aft. In exceptionally large steamers, such as the "Lusitania" and "Mauretania," four lines of shafting are arranged, with two ahead H.P. turbines and two ahead L.P. turbines, also two independent reverse turbines on the inner shafts. This arrangement, with the further addition of other two reverse turbines and two cruising turbines, is carried out in the case of large battleships and cruisers; sometimes the cruising turbines are compound,

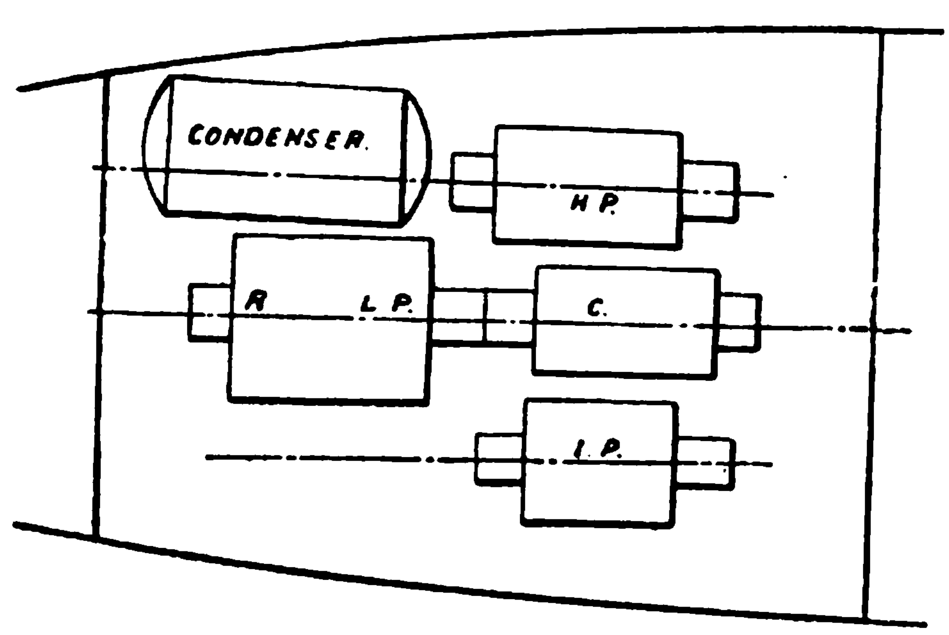
(1) **Standard arrangement,** one H.P. and two L.P. turbines, three shafts, one propeller to each.



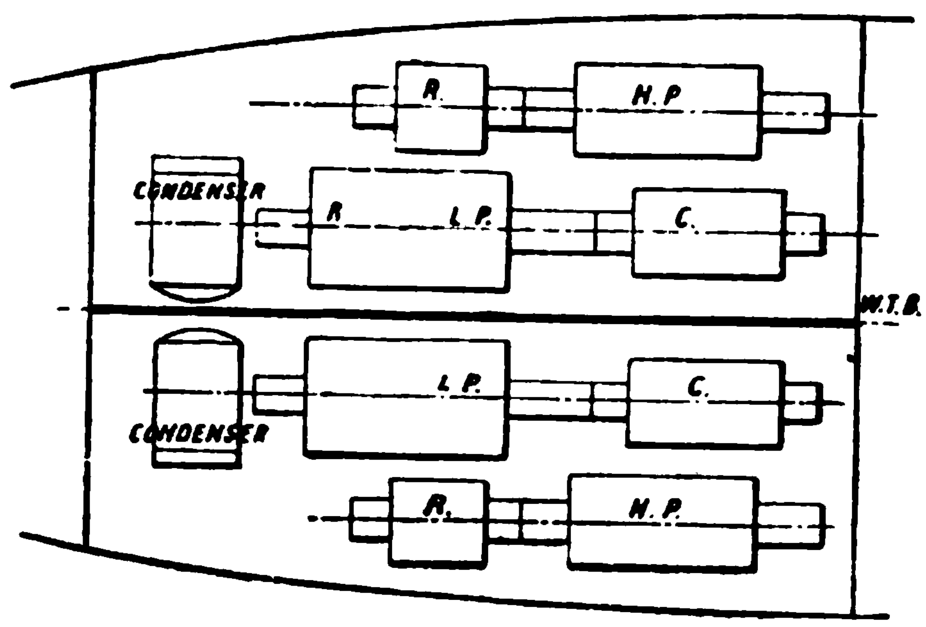
(2) **Large passenger steamer arrangement,** two H.P., two L.P., and two independent reverse turbines, four shafts, one propeller to each.



(3) **Torpedo craft arrangement,** one cruising, one I.P., one H.P., and one L.P. turbine, three shafts, one propeller to each.



(4) **Battleship or cruiser arrangement,** two cruising ahead, two H.P. ahead, two L.P. ahead turbines, also two H.P. reverse, and two L.P. reverse turbines, four shafts, one propeller to each.



one H.P. and one M.P., but generally both are of the same size, and receive direct steam from the boilers simultaneously. It should be noted that the Admiralty have decided to discard cruising turbines altogether in future, as in most cases the consumption at the low powers developed by these turbines does not justify their existence, in addition to the loss of power produced by the turbine blade resistance when running ahead or astern with the main turbines. Cruisers of the "Inflexible"-"Indomitable" type have ten turbines fitted, four ahead turbines—two H.P. and two L.P.—and four reverse—two H.P. and two L.P.—also two cruising turbines fitted, one on each H.P. turbine shaft, and intended for low cruising speeds and powers.

In torpedo craft the three-shaft arrangement is often carried out, but the turbines are arranged in triple series, one H.P. (centre), one M.P. (wing), and one L.P. (wing). Sometimes cruising turbines are fitted in addition to these in the case of large high-speed destroyers. The foregoing are the arrangements of turbines in present practice, but other arrangements have been proposed by the Parsons Company.

As regards the new combination arrangement of reciprocating engines and turbine, the steamers at present under construction are fitted with two wing triple or quadruple engines, both exhausting at a pressure below the atmosphere into the turbine on the centre shaft. An alternative design consists of one centre reciprocating engine exhausting into two wing turbines.

Steam Flow through Turbines.—In the standard turbine arrangement of five turbines—three ahead and two reverse—the steam, after expanding through the H.P. turbine, exhausts to both L.P. turbines simultaneously, and then to the two condensers. In the "Lusitania" design, the steam expands through each H.P. turbine, then through each L.P. turbine to the condensers of each respective side.

In the "Inflexible"-"Indomitable" class turbine arrangement, at full ahead power, the steam expands through each H.P., then each L.P., and then to the condensers of each side. At reduced ahead power, the steam first expands through each cruising turbine, then through each H.P. and L.P. turbine of each side, finally exhausting to the condensers. In running astern the steam first expands through the H.P. reverse turbine, then the L.P. reverse turbine, and finally exhausts to the condenser.

In the destroyer triple arrangement at full power, the steam first expands through H.P. turbine, then M.P. turbine, and L.P. turbine to the condenser, and at reduced power or cruising speed, the steam first expands through the cruising turbine, H.P. turbine, M.P. turbine, and L.P. turbine to the condenser.

Increase of Steam Volume.—To allow of the steam increasing in volume, as fall of pressure takes place, the various sets of blades increase in length from the forward to the after end, the clearance spaces between the blades also increasing in proportion, which necessitates packing pieces of a larger size being employed. The

blades also vary in shape or curvature, being flatter in section aft than forward. Each set of blades for each expansion requires its own allowance for expansion of metals by heat, so that the working clearance between the blades and casings or drums increases slightly throughout the turbine from forward aft. One of the practical difficulties met with in turbine construction at present is the correct adjustment for this expansion, as slight mishaps have occurred in one or two instances owing to fouling of the parts when heated up, the clearance allowance being insufficient.

Strictly, each successive ring of blades should be either of a wider pitch or greater height than the preceding one, as the steam is continuously falling in pressure and expanding in volume, but the Hon. C. A. Parsons considers the present arrangement quite near enough for practical purposes for all the difference that results.

NOTE.—The various extracts which follow, are from a paper read at the Forty-ninth Meeting of the Institution of Engineers and Shipbuilders in Scotland, on 24th October 1905, and entitled "The Determination of the Principal Dimensions of the Steam Turbine, with Special Reference to Marine Work," by Mr E. M. Speakman, Associate Member, &c., and are reprinted here by kind permission of the Council of that Institution. Describing the action of steam in passing through the successive rings of blades, &c., Mr E. M. Speakman says:—

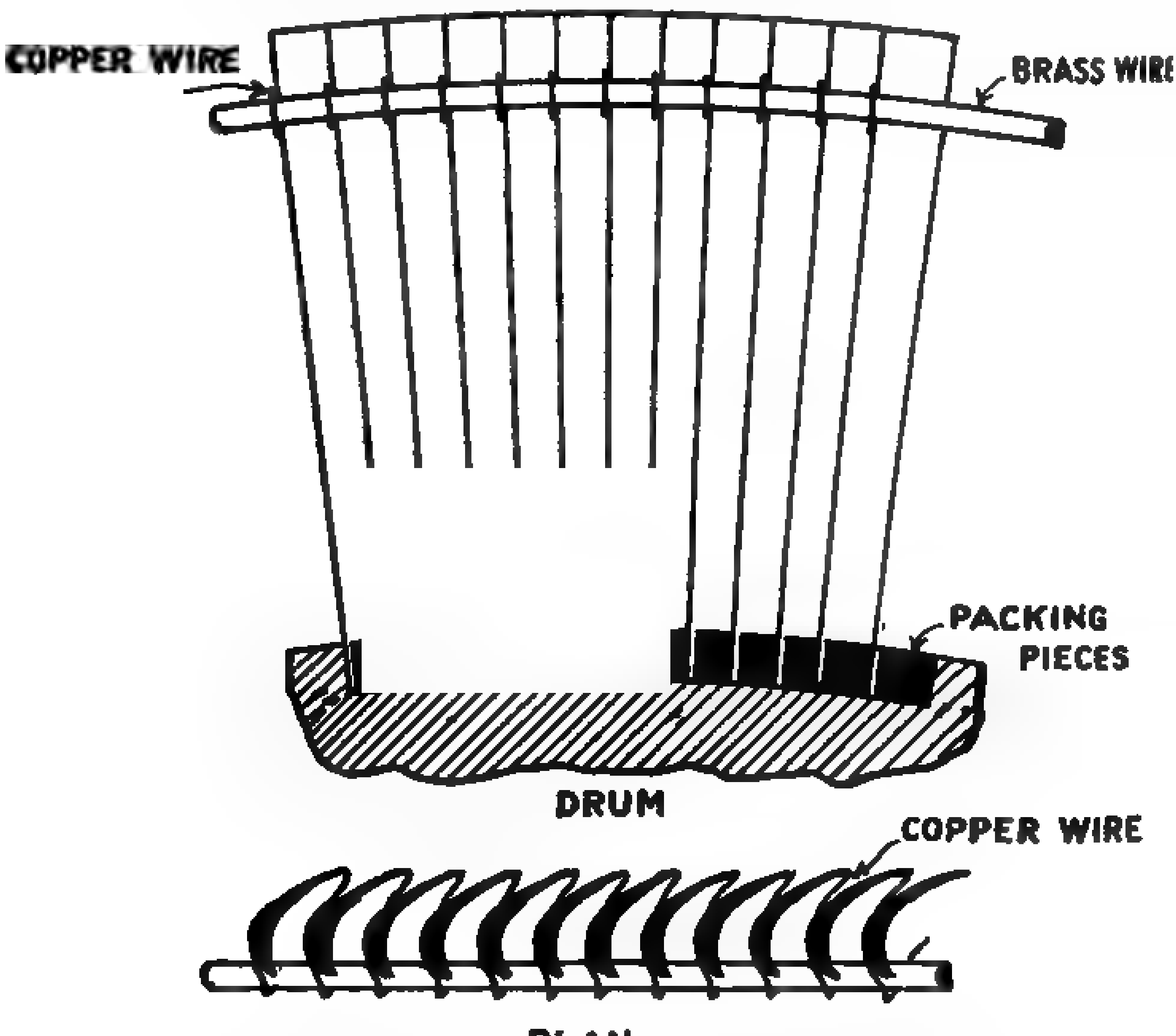
"Restricting attention to the design of the Parsons type of turbine, a few notes on the action of the steam among the blades may be of interest. Expanding through a definite range of temperature and pressure, steam exerts the same energy, whether it issues from a suitable orifice or expands against a receding piston. Two transformations of energy take place in the steam turbine—first, from thermal to kinetic energy; secondly, from kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine, the radical difference between the two lying in the low density of steam compared with water, and the wide variation of its volume under different temperatures and pressures.

"Fig. 5 gives a sectional elevation of a marine-turbine blading arrangement, and though this is only for an H.P. cylinder the principle is exactly the same throughout. The expansion, which is approximately adiabatic, is carried out in this annular chamber from A to B, which essentially resembles a simple divergent steam nozzle, but with this difference, that whereas in a nozzle the heat energy of the working steam is expended upon itself in producing high velocities, in Parsons' turbine the total expansion is subdivided into a number of steps, in each of which a certain dynamic relationship between jet and vane is maintained. The expansion of steam at any one stage is typical of its working throughout the turbine. Each stage consists of a ring of stationary blades which give direction and velocity to the steam, and a ring of moving blades that immediately convert the energy of velocity into useful torque. The total torque on the shaft is due to the impulse of steam entering the moving blades and to reaction as it leaves them, this process being repeated throughout the turbine.

"Leakage past the revolving portion of the spindle at D is almost entirely prevented by the ingenious form of frictionless packing, shown on a larger

scale on Fig. 6. The fine clearances and the sudden increase of section have the effect of alternately wire-drawing and expanding the steam, so that at successive grooves it becomes increasingly difficult for the steam to leak past the fine clearances. In the astern turbines, a radial form of packing, depending on fine tip clearances, must be adopted owing to the difference in expansion between spindle and cylinder. Numerous varieties of these forms of packing exist, some of them being extremely efficient in their action.

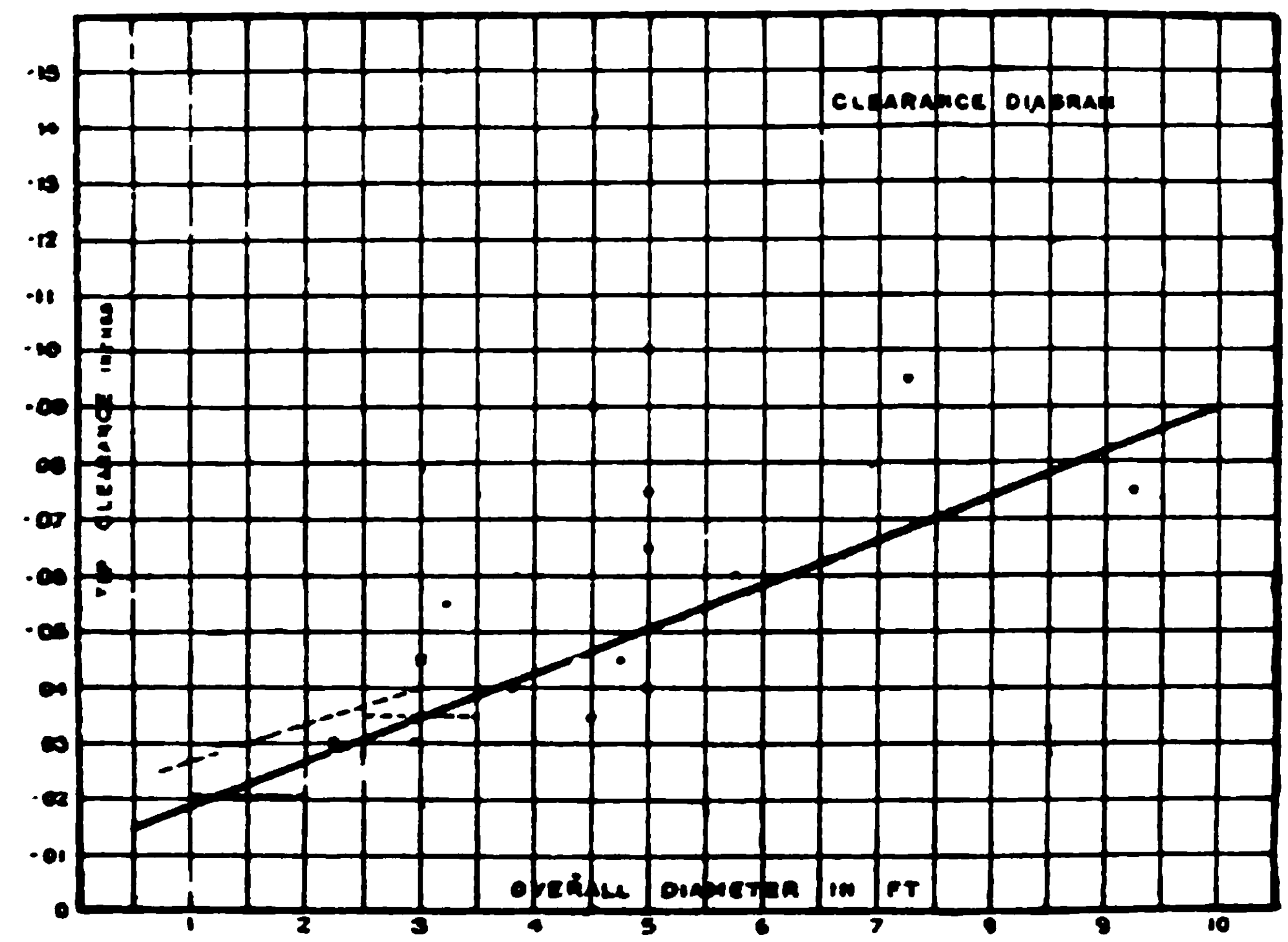
"The laws governing the best theoretical velocity of steam and blades are



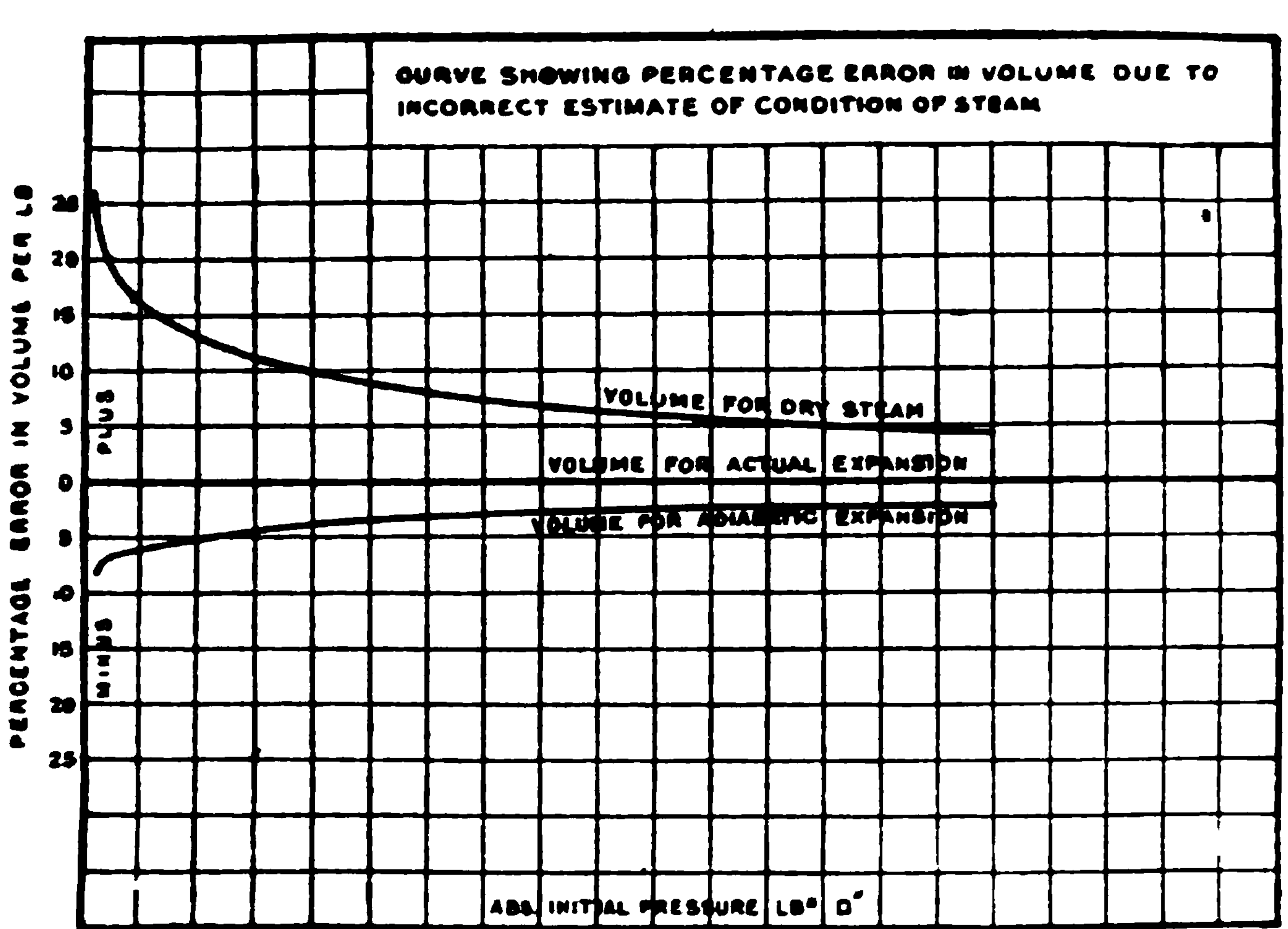
Elevation and Plan of Rotor Blades in Position, showing how secured (full size).

similar to those for water turbines, but in practice some modification is necessary, and the best ratio of blade speed and steam speed is still a matter of opinion. The ideal condition for impulse turbines occurs when the peripheral velocity of the buckets is one-half that of the jet, or in reaction turbines, when it is equal to it.

"Parsons' turbines, however, have been built with V_t, Fig. 7, varying from .25 to .85 of V_s, where V_t represents blade velocity at mean diameter, and V_s the steam speed due to expansion across the row in question. A very usual



Tip Clearance Diagram.



Correction Factors for Condition of Steam.

ratio in electrical work for large units has been $\frac{V_t}{V_s} = 0.6$, but this involves a greater number of rows than is possible in marine work, and the ratio must be reduced. These ratios need very careful calculation. The steam consumption must be accurately known in order to proportion them correctly throughout the turbine, and the necessity (which is inevitable with the present form of caulking piece) of having the same area of openings in so many rows while the steam volume increases so rapidly that it adds to the difficulty of close calculation.

"The potential energy of the steam, corresponding to the 'head' in water turbines, can easily be calculated for given pressure differences.

$$\text{B.Th.U.} \times 778 = \text{Energy in foot-pounds per pound of steam} = e.$$

$$V_s = 8\sqrt{e} = 223\sqrt{\text{B.Th.U.}}\ *$$

"For a given blade velocity, it is obvious, then, that the speed ratio between jet and vane must affect the number of stages, and the greater the ratio of V_t to V_s the greater will be the required number of rows, that is, to obtain the required V_s at each stage a smaller pressure drop per row is necessary, or *vice versa*.

"The best blading arrangement, scientifically and commercially, is the result of much theory and practice. The mean diameter is an arbitrary dimension capable of wide variation without affecting the efficiency, provided that the number of rows is correct; it is found by assuming, from experience, a blade velocity, whence—

$$\text{Mean diameter in inches} = \frac{\text{Blade velocity in feet per sec.} \times 228.\dagger}{\text{R. P. M.}}$$

"To arrive at the corresponding number of rows, the revolutions being given, the ratio of V_t to V_s must be settled, from which the steam speed can be obtained; it is a convenient assumption at the beginning of any design to consider the turbine as parallel throughout and of constant efficiency, and to design on this basis. The number of rows N on one diameter can be found by working out the B.Th.U.'s necessary to give a certain steam speed at each row, see Fig. 8, the available energy divided by the energy it is desired to abstract at each row will give the number of rows required. This result may be arrived at by various ways, but the principle involved is the same in each case. Numerous empirical coefficients for approximating steam speeds and the corresponding number of rows are obtainable from experience, and are similar in use and value to the Admiralty coefficient, that is, while they represent a crude method of doing something that should be done more scientifically, they are very simple and capable of rapid handling. Being, however, based on long and costly experiments, much reticence is observed regarding their publication. Varying, of course, with the steam pressure and vacuum, the number of rows on one diameter would involve an excessive length of turbine and also inconvenient blade heights. It is, therefore, usual to divide the rotor into three or more stages, which have the advantage of shortening the turbine and reducing the number of rows. If n = the fraction of power developed in

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* Constant $223 = \sqrt{64 \times 778}$.
NOTE.—64 = gravity × 2; 778 foot-pounds = 1 B.T.U.
† Constant $228 = (60 \times 12) \div 3.1416$.
NOTE.—60 seconds = 1 minute 12" = 1 foot.

View of Blades looking down on Rotor.

NOTE.—The scale of inches shown gives a fair idea of the blade widths and the distance apart of the various rows.

[*To face page* 22.

the first cylinder or barrel, $\frac{N}{n}$ = number of rows in the first barrel, and with the alteration of diameter and increase of blade velocity in the succeeding stages, the number of rows on other barrels are so altered as to keep, for equal powers and efficiencies—

$$(\text{Blade velocity})^2 \times \text{No. of rows} = \text{Constant.}^*$$

"The vane speeds adopted in practice vary considerably; for some time 100 feet per second was regarded as a standard for the first row, and I think the Westinghouse Company at Pittsburg was first to make a radical departure in this and adopt far higher speeds. The maximum vane speed used for Parsons' blading is, as far as the author is aware, about 375 feet per second in the low pressure blades, and 170 in the H.P. blades of electrical turbines; the lowest speeds used are in marine work, and are only about one-third of these. To some extent blade speed is governed by blade height; the speed should be so modified that this may be at least 3 per cent. of the mean diameter to reduce the proportion of clearance losses. Leakage over the tips of the blades is perhaps not so detrimental on account of actual leakage loss as in its superheating effect on steam between the row past which it leaks and the last row, because this reheating effect upsets calculations regarding openings by increasing the steam volume, and thereby affects the fluid efficiency. This leakage over the tips must be taken into account in designing reaction turbines. Temperature and diameter influence the clearance, and the stiffer the cylinder is to resist distortion due to heat the less it may be made. . . .

"TABLE II.—MARINE WORK.

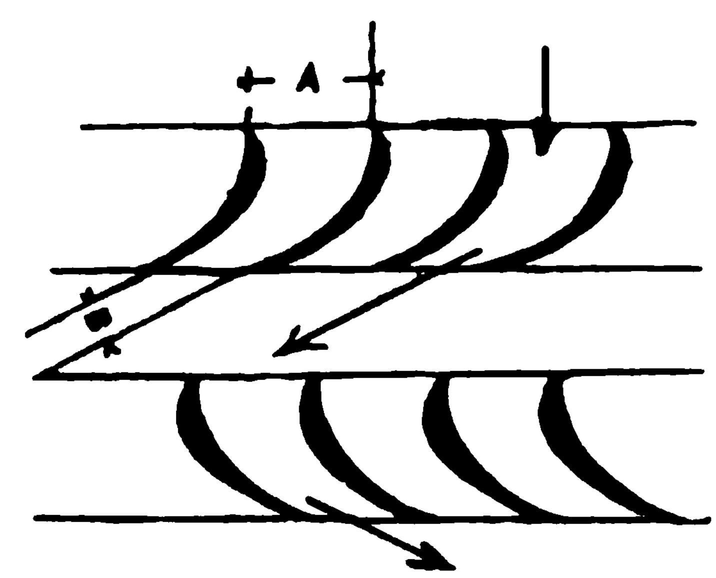
TYPE OF VESSEL.	Peripheral Vane Speed per Second.		Mean Ratio of V_t—V_s.	Number of Shafts.
	H.P.	L.P.		
High speed mail steamers - -	70- 80	110-130	·45-·5	4
Intermediate do. - - - -	80- 90	110-135	·47-·5	3 or 4
Channel steamers - - -	90-105	120-150	·37-·47	3
Battleships and large cruisers -	85-100	115-135	·48-·52	4
Small cruisers - - - - -	105-120	130-160	·47-·5	3 or 4
Torpedo craft - - - - -	110-130	160-210	·47-·51	3 or 4

"In Table II., the vane speeds adopted in various classes of work are given, and the reduction in peripheral speed on account of the propeller reducing the revolutions, and the necessary proportion of blade height modi-

* NOTE.—The "constant" referred to above appears to vary from 1400000 to about 1600000.

fying the diameter may be clearly seen. To this combined action is due t fact that only in the faster classes of vessels, or in those small types in wh some propulsive efficiency can be sacrificed, is the turbine applicable. slow cargo steamers, though the revolutions may be high enough, the po required is not sufficient to enable a reasonable blade height to be adopt and it is this consideration—viz., proportion of leakage over blade tips—t curtails the wider adoption of this type of turbine. For the same low periphe- blade speed, other types of turbine are unsuitable on account of the impos bility of reducing the steam velocity sufficiently without abnormal weight a inefficiency.

"The smallest size of marine turbine is usually larger than the aver electrical turbine as far as power is concerned, and therefore does not m with the same commercial considerations as the smaller sizes of the latter ty These are not designed for the same internal efficiency as the larger machi



A = INLET

B = OUTLET

Diagram showing Area through normal Blades to allow passage of Steam, compared to Blade Annulus Area.

B = A ÷ 3.

The area B depends on the steam velocity and steam volume, being more for low-pressure steam than for high-pressure steam.

chiefly on account of manufacturing cost, and they do not attain anything l the same efficiency compared with the Rankine cycle.

"Speaking in reply to the discussion on his paper to the Institution Naval Architects in 1903, Mr Parsons said that, 'for all practical purpo while the steam is traversing each set (of blades) as shown, it behaves like incompressible fluid, just like water would do, as the expansion is very sm at each set. The frictional losses and the eddy-making losses would be pra cally identical within small limits with what they would be with water, and t actual forces would be in proportion to the density of the medium. . . . the turbine blades themselves, the efficiency is between 70 and 80 per cent.'

"Using this hydraulic analogy enables us to calculate the number of sta required in a different manner: the 'equivalent head,' due to the steam pressu(es), may be found, together with that at each row necessary to give the requir velocity, from which both the number of stages and the coefficient of expansi at each stage may be worked out.

TO MARINE WORK.

APT.

STEAM SPEED FEET PER SECOND

4

"In the early marine designs, such as the 'Queen Alexandra' and H.M.S. 'Amethyst,' the turbine drums were all made of the same diameter, and the higher speed necessary on the L.P.'s was got by running at considerably higher revolutions than on the H.P. shaft; but, following up the increase in propeller efficiency found to be due to the use of larger screws, the speed for each shaft is now more nearly equal, while the wing drums are made larger in diameter. The vagaries of the following wake, however, necessitate slightly different propeller dimensions on each shaft, or else slightly different revolutions with the same screws; and it is noticeable that in a triple-screw arrangement, the centre screw being right-handed and the wing screws revolving outwards, that the starboard propeller is influenced by the centre one, and almost invariably revolves at a lower speed. In a four-shaft design, due to the varying wake values at different speeds, and possibly, also, to some unequal distribution of power, the outer screws run slower at low speeds and faster at high speeds than the two inner shafts, but exact data as to this, and the possibility of allowing for it in the design, are still wanting.

"In all types of turbines—Parsons', Rateau's, Curtis', &c.—a certain ratio must be maintained between the blade velocity and steam velocity, and as steam acquires very high velocities by expansion, the blade velocity must be maintained either by the revolutions or by large diameters, or both. As the weight increases very rapidly with the diameter, and extraordinarily so with the reduction in rotative speed, it is preferable to increase, if possible, the revolutions or the number of stages rather than the diameter, and especially should this be done in cases where, as in the Rateau or Zoelly types, the weight increases more rapidly in inverse proportion to the R.P.M. and the diameter than it does with other types. To increase the revolutions, it may be necessary to increase the number of shafts and propellers, thus reducing the power per shaft and the effective thrust through each screw. Increasing the diameter of the turbine adds largely to the constructional difficulties, especially of the cylinder.

"Having obtained the number of rows and the diameter, the blading arrangement can be worked out in detail. The height of blade depends on the volume of the steam and the speed at which it is to flow, and also on the ratio of the area of exit openings between the blades to that of the annulus between spindle and cylinder, which is about one-third in normal blades. The necessary clear area to pass the steam being equal to volume ÷ velocity, and knowing this annular factor, say 3, for a ratio of one-third (or 2 for $\frac{1}{2}$, &c.), then

$$\text{Height of blade in inches} = \frac{\text{Clear area in square inches} \times 3}{\text{Mean circumference in inches}}.$$

"The ratio of blade height to mean diameter should not be less than 3 per cent. or more than 15 per cent., because in the former the leakage will be excessive, and in the latter the bending moment on the blade becomes too great, and the radial divergence of the blades too much. The width of blade, the shape of section adopted, and the circumferential pitch, are standard considerations, and affect the factor 3 given above. It is not proposed to enlarge upon them in this paper. It may, however, be remarked that for $\frac{V_t}{V_s}$ greater than .6 the usual shape of Parsons' section, as shown in Fig. 5, should be modified to a somewhat different form of blade, with a sharper entrance edge. This section is not to be recommended, as, owing to the necessity of

stiengthening the blade sufficiently, the metal must be placed nearer the exi edge thus increasing the angle between the face and the back of the exit edg of the blades, and giving, in fact, an inferior shape of opening compared witl that obtainable with a blade section adapted to ratios under .6. If, for th present, it is sufficient to use the blade sections and packing pieces similar t those now adopted so generally, in Table III. can be found a list of widths fo a given height, and the axial spacing of the rows. While this must be kep down to reduce the length of drum, it must be sufficient to allow for some pla in overhauling; and sufficient clearance can be allowed here without affectin the economy. The openings between the blades to allow of the passage o the steam are very important, and must be carefully designed. The actu volume of the steam—not the volume per lb., as found in tables, or the volum due to adiabatic expansion, but the exact volume per lb. at any point alon the turbine—must be determined, in order to arrive at the desired adjustme of velocities. It is extremely doubtful whether the present blading arrang ments give the best results; greater accuracy of calculation, and consequentl improved pressure distribution and efficiency, seem likely to follow the use o a more mechanical blading construction."

Revolutions and Rotor Diameter.—The highest theoretic efficiency of the turbine is attained when the linear velocity of th rotating blades is about equal to one-half the velocity of the stea impinging upon those blades, and as this figure is very high the remain but two alternatives to obtain the results which the be efficiency would require. The first of these two is to arrange th revolution speed so high as to enable the vanes to receive the stea under the conditions stated above. The other alternative is to redu the speed of rotation of the turbine by increasing its diameter in equ ratio to the reduction of rotative speed, which, of course, has the di advantage of increasing the weight of turbines required.

Theoretical Blade Heights.—The next diagram shows how th blade heights would vary if made to exactly correspond to the stea expansion.

As the pressure falls the volume increases (adiabatically), so th each succeeding row of blades should be slightly longer than th preceding row. In practice, however, this is not carried out, th blades being arranged in sets of equal height, or, as it is calle "stepped." The horizontal dotted lines show the actual arrangeme of blade heights as usually fitted. The 6th, 7th, and 8th expansio of each L.P. turbine consist of rows of blades of equal height, but should be noted that the angle or curvature of each set is differen the 7th and 8th expansions having "wing" blades of flatter sectio and of wider circumferential pitch than those of the preceding sets.

It will be observed that the first expansion of the L.P. has *shorte* blades than the last expansion of the H.P. It must, however, be remembered that the diameter of the L.P. drums is *more* than the H.P., thus giving a higher peripheral and steam speed, also that in the usual standard arrangement of one H.P. and two L.P. turbines only *half* the quantity of steam passes through each L.P. turbine.

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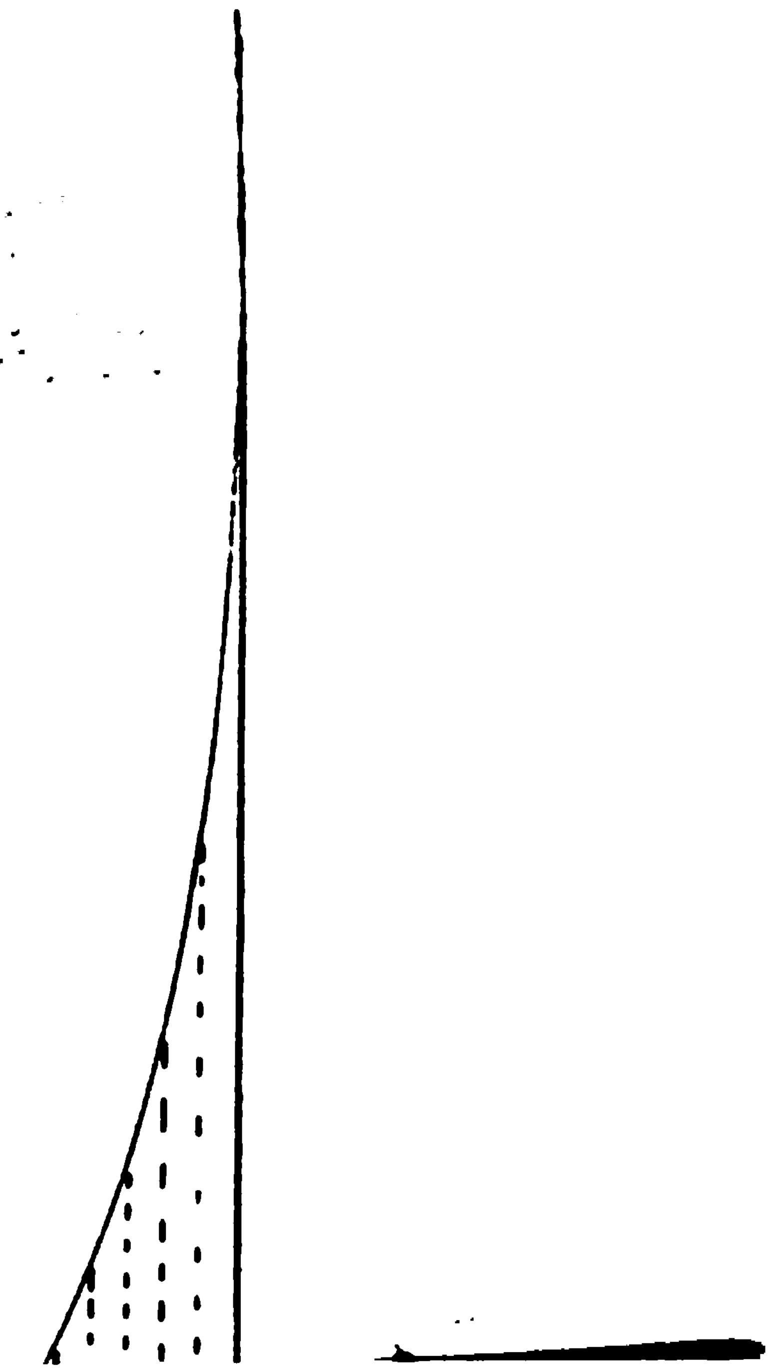

the increasing volume of steam passing through the last section. I the last set of blades is made up of, say, 24 rings, all of the same height, then 8 rings are as of the 1st set, 8 of the 2nd set, and the remaining 8 as of the 3rd set. Observe that the blades are less curved aft, also that the pitch is increased, which results in a smaller

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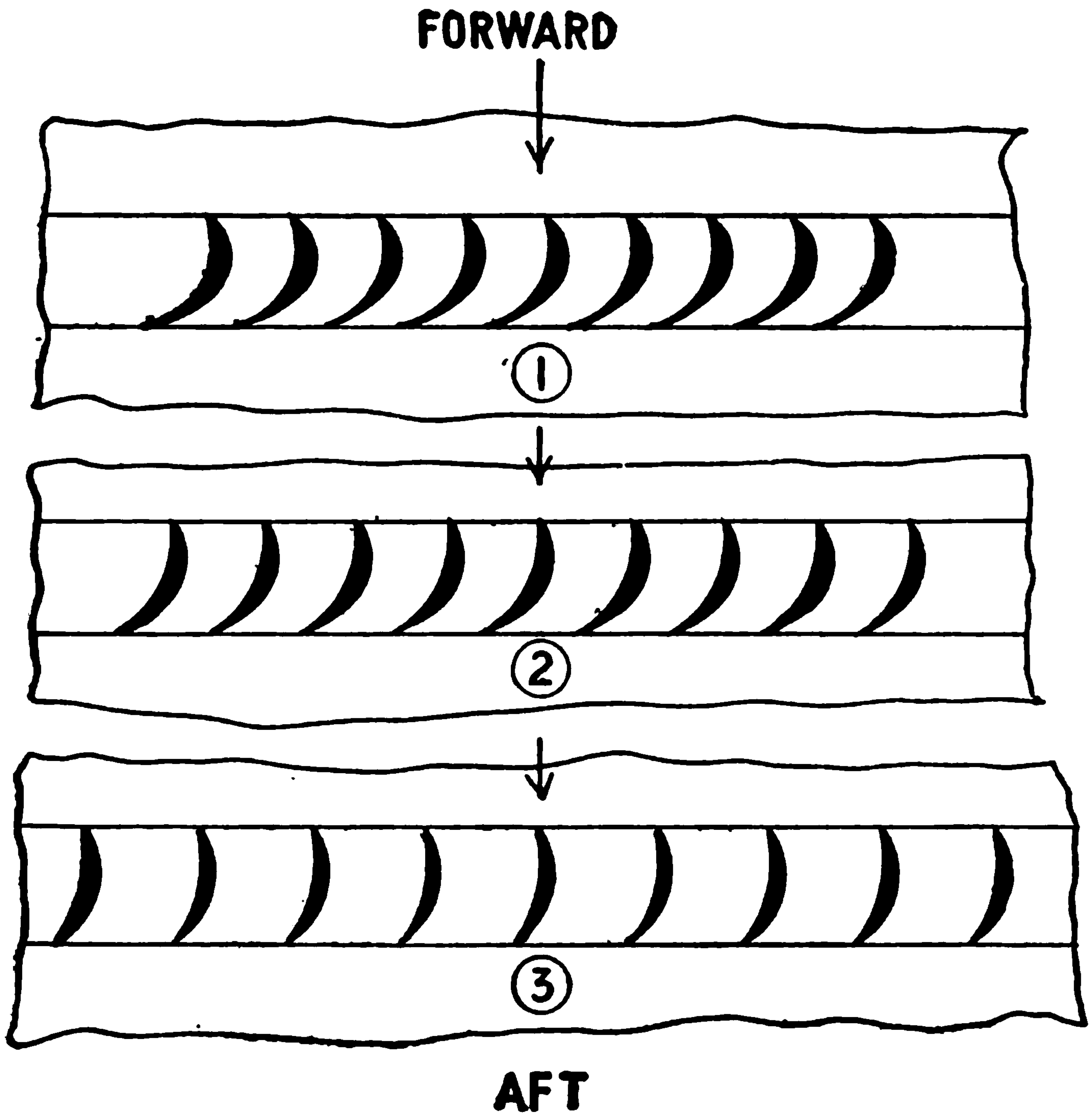
NOTE.—Observe that the total number of blades is generally the same for the H.P. turbine and each L.P. turbine, the H.P., however, only having four expansions, each one containing *twice* the number of blades contained in each of the eight expansions of the L.P. turbines.

Expansion Clearance.—At each change of expansion a clearance space is arranged between the last ring of blades of one set, and the first ring of the next set (see sketch) to allow of drop of pressure.



Variation in Blade Angles and Pitch in last three L. P. Expansions.

1. Sixth Expansion. 2. Seventh Expansion. 3. Eighth Expansion.

NOTE.—The above are examples of "wing" blades.

Blade Variation.—In the largest size of blades which are fitted on the after or low pressure end of each turbine, the curve and pitch is varied so as to really constitute three expansions, although the *height* of the blades is the same. The sketch above shows the variation of blade curvature and pitch, which is arranged to allow for the increasing volume of steam passing through the last section. If the last set of blades is made up of, say, 24 rings, all of the same height, then 8 rings are as of the 1st set, 8 of the 2nd set, and the remaining 8 as of the 3rd set. Observe that the blades are less curved aft, also that the pitch is increased, which results in a smaller

number of blades per ring. Theoretically each successive ring of blades throughout the whole turbine should be graded in this way.

NOTE.—In some cases the exit openings of the first half of the rows of each of the first few expansions have been reduced, which practically gives two expansions to what was formerly only one: this alteration increases the velocity of the steam passing between the rows with the restricted openings. A specially designed "closing-up" tool is employed to reduce the blade openings as described.

Velocity Calculations, &c.—The Foot-pounds of energy contained in a given weight of steam at a given pressure and velocity are found as follows:—

$$\text{Kinetic Energy} = \frac{W \times V^2}{64.4} \text{ Foot-pounds.}$$

NOTE.—W = Weight of steam in pounds.
V = Velocity of steam in feet per second.
64.4 = Acceleration due to gravity per sec. per sec.

The change in kinetic energy or the work done between the blades may be expressed as follows, friction and other losses, such as tip clearance leakage, being neglected.

$$\text{Kinetic Energy} = \frac{(V^2 - v^2) \times W}{64.4} \text{ Foot-pounds.}$$

Where V = Velocity of steam in feet per second at the entering edge of blades.
v = Velocity of steam in feet per second at the leaving edge of blades.
W = Weight of steam in pounds.
64.4 = Acceleration due to gravity per sec. per sec.

For example—Let V = 600 feet per second.
" W = 1 pound.

$$\text{Then, } \frac{W \times V^2}{64.4} = \frac{1 \times 600^2}{64.4} = 5590 \text{ Foot-pounds of kinetic energy.}$$

For example—Let V = 400 feet per second.
" v = 300 " "
" W = 1 pound.

$$\text{Then, } \frac{(V^2 - v^2) \times W}{64.4} = \frac{(400^2 - 300^2) \times 1}{64.4} = \begin{cases} 1087 \text{ Foot-pounds in passing} \\ \text{across one row of blades.} \end{cases}$$

NOTE.—In turbine practice the steam expands approximately adiabatically. No heat being supplied from without, and if no leakage of heat takes place, the work is done at the expense of the internal heat energy of the steam, and the fall in pressure, in temperature, and amount of condensation in the turbine is proportional to the work done.

The expansion being adiabatic, the heat drop for a given pressure drop is much more than that shown in the total heat table of saturated steam, as the steam which condenses during the performance

of work reduces the weight of actual steam remaining after expansion, and therefore the heat contained per pound of steam and water mixture is proportionally less. It should be again noted that the internal heat energy of the steam is transformed into mechanical energy, hence the heat drop so often referred to.

Let V = Velocity of steam.
„ W = Weight.
„ H = Heat units given up.
64.4 = 32.2 × 2 (acceleration due to gravity per sec. per sec).

$$\text{Then, } \frac{V^2 \times W}{64.4} = \text{Foot-pounds of kinetic energy}.$$

$$\text{And, Foot-pounds} \times 64.4 = V^2 \times W.$$

$$\text{Therefore, } H \times 778 \times 64.4 = V^2 \times W.$$

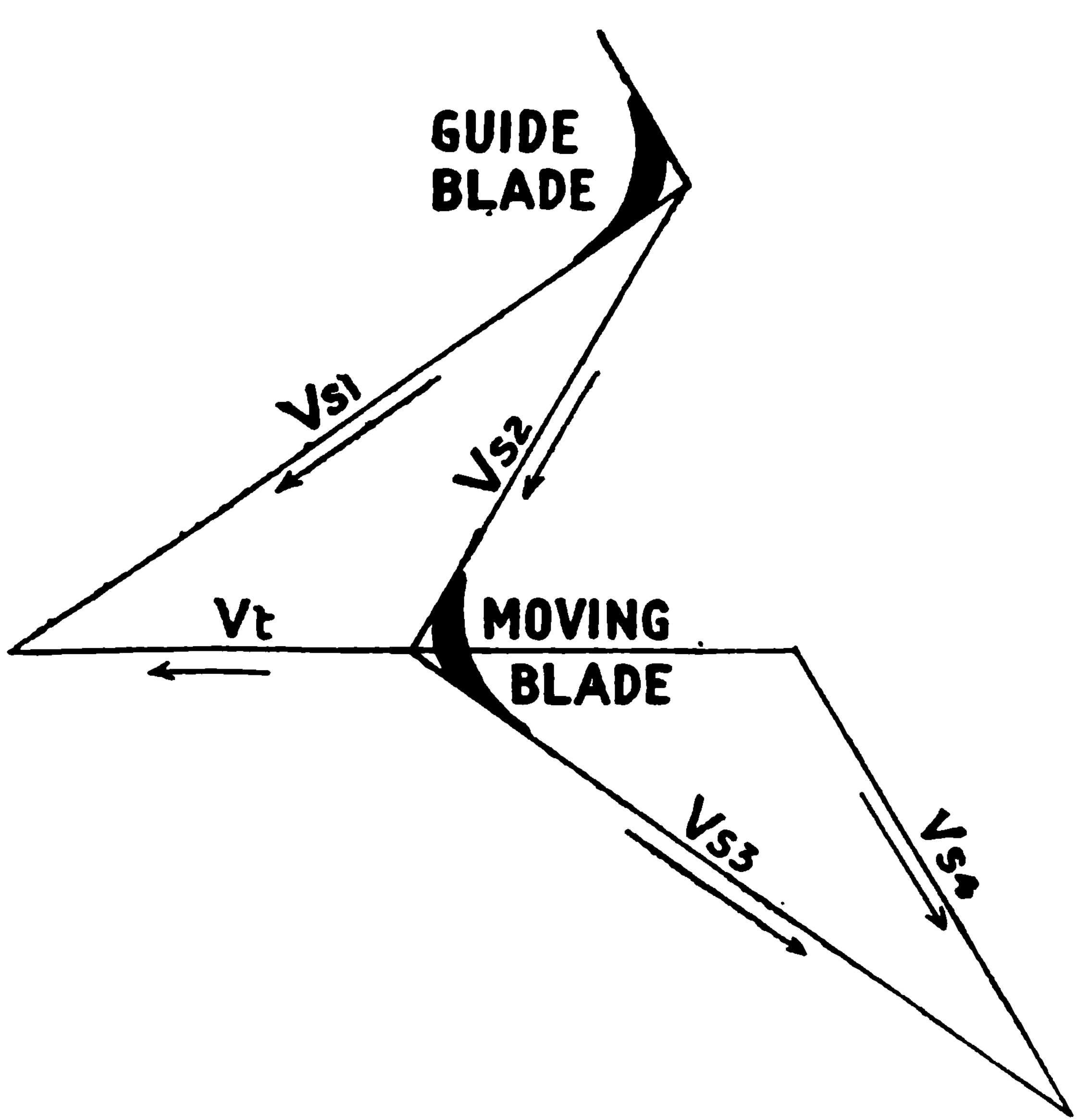
$$\text{,,} \quad V^2 = \frac{H \times 778 \times 64.4}{W}$$

$$\text{,,} \quad V = \sqrt{\frac{H \times 778 \times 64.4}{W}}$$

$$\text{,,} \quad \frac{V^2 \times W}{64.4} = H \times 778.$$

$$\text{And, } \frac{\text{Foot-pounds}}{778} = H.$$

Velocity Diagram.



The above diagram shows graphically the varying steam velocities previously referred to.

V_{S1} = absolute steam velocity (initial).
V_{S2} = relative „ „ „ with regard to blade or rotor speed.
V_{S3} = relative „ „ leaving the blades.
V_{S4} = absolute „ „ „ „
V_t = blade velocity.

The actual velocity of the steam passing between the rows of blades depends upon the drop of pressure produced by the expenditure of Foot-pounds of heat energy developed as work done.

The velocity of the steam in feet per second, due to any pressure drop, is found as follows:—

$$\text{Velocity of steam} = \sqrt{64.4 \times 778 \times (H_1 - H_2)}.$$

NOTE.—H_1 = Initial heat units.
H_2 = Final „ „
64.4 = twice 32.2 (gravity acceleration per sec. per sec.).

EXAMPLE.—Calculate the velocity of the steam between the blades of a Parsons turbine if the heat units per pound of steam at admission edge are 1,200, and at exhaust edge 1198 units (a heat drop of 2 units, neglecting frictional and other losses).

$$\text{Then, } \sqrt{64.4 \times 778 \times (H_1 - H_2)} = \sqrt{64.4 \times 778 \times (1200 - 1198)}$$
$$= 316 \text{ ft. Velocity per second.}$$

EXAMPLE.—Calculate the energy contained in one pound of the above steam.

RULE.—$\dfrac{V^2 \times W}{64.4}$ = Foot-pounds; Therefore, $\dfrac{316^2 \times 1}{64.4}$ = 1556 Foot-pounds, and $1556 \div 778 = 2$ B.T. Units as before.

Absolute Velocity and Relative Velocity.—By absolute velocity is meant the velocity of the steam with regard to stationary objects.

By relative velocity is meant the velocity of the steam with regard to moving objects, in this case with regard to the blade velocity.

The two steam velocities referred to are shown graphically in the sketch, and it will be observed that the rotor velocity affects the relative velocity of the steam inversely, that is, with high rotor speed the relative steam speed on leaving the moving vanes is low, and *vice versa.*

Guide Blades.—In the guide blades, the work done in producing increase of steam velocity is equal to the following:—

$$\frac{V_{S1}^2 - V_{S4}^2}{64.4} = \text{Foot-pounds in guide blades.}$$

Moving Blades.—In any of the moving blades, the work done in producing shaft rotation is equal to the following:—

$$\frac{V_{S3}^2 - V_{S2}^2}{64.4} = \text{Foot-pounds in moving blades.}$$

Notice that the work done in the moving blades is calculated from the *relative* steam velocity of admission and *relative* steam velocity of exit, whereas the work done in the guide blades is calculated from the absolute steam velocities of both admission and of exit.

Work of Steam Acceleration.—The work done by the steam upon itself in the way of increased velocity in the first row of stationary or guide blades of a turbine is, as explained above, found as follows :—

$$\frac{V^2 \times W}{64.4} = \text{Foot-pounds.}$$

EXAMPLE.—Calculate the Foot-pounds of energy developed by one pound of steam within the guide blades of a turbine if the steam velocity is 300 feet per second.

$$\text{Then, } \frac{V^2 \times W}{64.4} = \frac{300^2 \times 1}{64.4} = 1400 \text{ Foot-pounds (nearly).}$$

In any turbine, then, the total energy per pound of steam supplied is equal in Foot-pounds to the following :—

$$(1114 + .3 \times T^\circ) \times 778 = \text{Foot-pounds contained.}$$

Therefore, after the steam has passed through a pair of rows of blades, the equation will be as follows :—

$$(1114 + .3 \times T^\circ) \times 778 = \frac{V^2}{64.4} + H \times 778.$$

Where H = heat left in the steam after fall in pressure between blade rows.

From the above it will be obvious that the total energy remains the same, and is exactly equal to the work done added to the energy still remaining in the steam, part of which will be represented by the small per cent. of water condensed, and the remainder by the actual steam velocity.

EXAMPLE.—At the inlet edge of the blades the pressure is 140 lbs., and at the outlet edge 138 lbs., the difference of B.T. units due to the nature of the expansion being $2.5 = (H_1 - H_2)$. Calculate (1) the steam velocity, (2) the kinetic energy of the steam.

$$V = \sqrt{64.4 \times (H_1 - H_2) \times 778}.$$

$$\text{Therefore, } \sqrt{64.4 \times 2.5 \times 778} = 354 \text{ feet per sec.}$$

$$\text{and, } \frac{V^2}{64.4} = \text{kinetic energy} = \frac{354^2}{64.4} = 1945 \text{ Foot-pounds.}$$

EXAMPLE.—The two velocities of the steam are $V_1 = 300$ feet per second, and $V_2 = 450$ feet per second. Calculate the Foot-pounds of work given up, and the number of B.T. units of heat drop across the blades per pound of steam.

$$\text{Kinetic energy} = \frac{(V_2^2 - V_1^2) \times W}{64.4} = \frac{(450^2 - 300^2) \times 1}{64.4} = 1746.8 \text{ ft.-lbs.}$$

$$\text{Heat drop} = \left(\frac{V^2 \times W}{64.4}\right) \div 778 = \left(\frac{450^2 \times 1}{64.4}\right) \div 778 = 4.04 \text{ units of heat.}$$

Notice that the relative velocity of the entering steam is less than its absolute velocity, and that the relative velocity of the steam leaving the moving blades is *more* than its absolute velocity. This is due to the velocity of the blades being in one constant direction.

Blade Velocity.—The mean velocity of the blades being often from .4 to .5 of the steam velocity, then the steam velocity $\times .5 =$ blade velocity at mean diameter of the blade circle. So that if the steam velocity is, as stated, 18000 feet per minute, then,

$$18000 \times .5 = 9000 \text{ feet per minute.}$$

Now if the revolutions are to be not more than, say, 400 per minute, the diameter of rotor can be found as follows :—

$$\frac{9000}{400 \times 3.1416} = 7.08 \text{ feet diameter across mean diameter of blades}$$

The actual diameter of rotor would probably be somewhere under 7 feet, when the length of blades is deducted.

Rotor Diameter.—The difference in turbine diameter required for low and high revolution speed is shown in the following cases :—

Let V_s = steam velocity, and V_t = blade velocity.

EXAMPLE.—Steam velocity $V_s = 21000$ feet per minute, and if $\frac{V_t}{V_s} = .5$, find the diameter of rotor required for (1) 600 revolutions, and (2) 200 revolutions per minute.

Then, (1) $21000 \times .5 = D \times 3.1416 \times 600.$

$$\therefore \frac{21000 \times .5}{3.1416 \times 600} = 5.5 \text{ feet diameter of rotor.}$$

Again, (2) $21000 \times .5 = D \times 3.1416 \times 200.$

$$\therefore \frac{21000 \times .5}{3.1416 \times 200} = 16.7 \text{ feet diameter of rotor.}$$

NOTE.—The above diameters are measured across the mean *height* of blades, so that the actual diameter of rotor might be, say, 9 or 10 inches less than the above results.

Increase of Velocities.—At each succeeding expansion, the mean diameter of the blades being increased owing to increase of blade height, the velocity of the blades per second at the after or exhaust end of any turbine rotor is rather more than at the admission or forward end. An example will make this clear.

EXAMPLE.—Calculate the respective blade velocities at each expansion of the H.P. turbine, if the blade heights are as follows:—

$1\frac{1}{16}$ in. at 1st expansion.
$1\frac{3}{8}$ in. „ 2nd „
$1\frac{7}{8}$ in. „ 3rd „
$2\frac{1}{2}$ in. „ 4th „

The diameter of rotor is 5 feet 6 inches, and the revolutions per minute 600. Diameter of rotor = 3 feet 6 inches = 42 inches, then

At 1st expansion, 42 in. + $1\frac{1}{16}$ in. = $43\frac{1}{16}$ in. = 43.0625 mean diameter across blades.

At 2nd expansion, 42 in. + $1\frac{3}{8}$ in. = $43\frac{3}{8}$ in. = 43.375 mean diameter across blades.

At 3rd expansion, 42 in. + $1\frac{7}{8}$ in. = $43\frac{7}{8}$ in. = 43.875 mean diameter across blades.

At 4th expansion, 42 in. + $2\frac{1}{2}$ in. = $44\frac{1}{2}$ in. = 44.5 mean diameter across blades.

Therefore, $\dfrac{43.0625 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{112 ft. per second velocity of blades}\\ \text{at 1st expansion.}\end{array}\right.$

„ $\dfrac{43.375 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{113 ft. per second velocity of blades}\\ \text{at 2nd expansion.}\end{array}\right.$

„ $\dfrac{43.875 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{114 ft. per second velocity of blades}\\ \text{at 3rd expansion.}\end{array}\right.$

„ $\dfrac{44.5 \times 3.1416 \times 600}{60 \times 12} = \left\{\begin{array}{l}\text{116 ft. per second velocity of blades}\\ \text{at 4th expansion.}\end{array}\right.$

NOTE.—Divide by 60 seconds and by 12 inches to obtain feet per second.

NOTE.—The blade tip clearance, about $\frac{50}{1000}$ of an inch, or .05 in. is neglected in the above calculations.

It should be noted that the steam velocities increase correspondingly at each successive expansion, so that the average ratio of blade velocity to steam velocity, or $\frac{V_t}{V_s}$, is maintained more or less constant throughout the turbine.

From the foregoing, it will be easily seen that the blade heights and openings must increase at a high ratio to allow for the very rapid increase of volume at the lower pressures carried near the exhaust end of the low-pressure turbines.

Steam Velocity.—The velocity or speed of steam at any given pressure varies according to the pressure of the exhaust or opposing back pressure. In marine turbine practice, the velocity of the steam is often about 300 feet per second, or 18,000 feet per minute.

For example, at an initial pressure of 180 lbs. the velocity of steam when flowing directly into the atmosphere is equal to about 3,000 feet per second, and if allowed to flow into a vacuum of 25 inches the velocity is about 3,700 feet per second.

After passing each row of rotor blades a small drop in pressure and in heat energy takes place owing to the useful work done by the steam in rotating the shaft and in increasing its own velocity. This drop is proportional to the foot-pounds of energy expended at each row of blades, and, as before stated, the exchange of heat into work or kinetic energy is equal to 778 foot-pounds per unit of heat. The average drop in pressure per row of blades throughout a marine turbine works out as about .76 of a pound.

Heat Drops.—The fall of pressure, temperature, and heat units in the casing of a turbine may be described in a somewhat rudimentary form as follows:—As previously explained, the necessary steam velocity required to produce effective work, or kinetic energy, on the blades of the turbine rotor is obtained by allowing a suitable drop in pressure and in heat units between the successive pairs of rows of blades. It is important to note that the average drop per pair of rotor blade rows is less in the L.P. turbines than in the H.P. turbine. This is accounted for by the fact that the difference in total heat units of steam at low pressure is more than that of high-pressure steam for the same pressure drop, or in other words, an equal difference in drop of heat units can be obtained by a smaller drop of pressure. The difference in drop of heat units necessary for the required foot-pounds of energy to be developed can therefore be obtained by a smaller pressure drop per pair of rows with low-pressure steam, as the following examples will perhaps make clear:—

Work done during Adiabatic Expansion.—To calculate the work done, or, which is the same thing, the units of heat given up or converted into work during the adiabatic expansion of steam in a turbine, the following data are required:—

The absolute temperature of the steam before and after expansion.
The latent heat of the steam before and after expansion.
The dryness factor of the steam before and after expansion.

Let T_1° = Absolute temperature before expansion.
T_2° = " " after "
H_1 = Latent heat before expansion.
H_2 = " " after "
f_1 = Dryness factor before expansion.
f_2 = " " after "

The heat energy given out in British Thermal Units

$$= f_1 \times H_1 - f_2 \times H_2 + T_1^\circ - T_2^\circ = \text{B.T.U.}$$

And, $\text{B.T.U.} = \dfrac{V^2}{2g \times 778}$; or, $V^2 = 2g \times 778 \times \text{B.T.U.}$

Therefore, $V = \sqrt{2g \times 778 \times \text{B.T.U.}}$

NOTE.—V = Velocity of steam in feet per second.
$2g = 2 \times 32.2$ = Acceleration due to gravity in feet per second per second.

NOTE.—The "Entropy" diagram affords the clearest explanation of the heat utilisation and expenditure.

EXAMPLE 1—**High-Pressure Steam.**—In a marine turbine find the heat energy given up by one pound of steam when flowing from one expansion where the pressure is 150 lbs. absolute, to another expansion where the pressure is 140 lbs. absolute, given that—

$$T_1^\circ = (358^\circ + 461^\circ) = 819.$$
$$T_2^\circ = (353^\circ + 461^\circ) = 814.$$
$$H_1 = 861.$$
$$H_2 = 865.$$
$$f_1 = 1.$$
$$f_2 = .996.$$

NOTE.—Absolute temperature = Fahrenheit + 461°.

$$\text{Heat units given up} = 1 \times 861 - .996 \times 865 + 819 - 814$$
$$= 861 - 861.54 + 819 - 814 = 1680 - 1675.5 = 4.5 \text{ B.T.U.}$$

Also, calculate the velocity of the steam.

Then, $V = \sqrt{64.4 \times 778 \times 4.5} = 474$ feet per second.

EXAMPLE 2—**Low-Pressure Steam.**—In a marine turbine find the heat energy given up by one pound of steam when flowing from one expansion where the pressure is 6 lbs. absolute, to another expansion where the pressure is 2 lbs. absolute, given that—

$$T_1^\circ = (170^\circ + 461^\circ) = 631^\circ.$$
$$T_2^\circ = (126^\circ + 461^\circ) = 587^\circ.$$
$$H_1 = 995.$$
$$H_2 = 1026.$$
$$f_1 = .85.$$
$$f_2 = .8.$$

$$\text{Heat units given up} = .85 \times 995 - .8 \times 1026 + 631 - 587$$
$$= 845.75 - 822.8 + 631 - 587 = 1476.75 - 1409.8 = 66.95 \text{ B.T.U.}$$

From the foregoing it will be seen that in Example No. 1, with a pressure drop of 10 lbs. (from 150 to 140 lbs.) and the conditions stated, the number of heat units converted into work, or kinetic energy, is only 4.5, whereas in Example No. 2, with a *pressure drop of only* 4 *lbs.* (from 6 lbs. to 2 lbs. absolutely), the number of heat units converted into kinetic energy is 66.95, thus clearly indicating the greater value of lower pressure steam in turbine practice, and accounting for the two L.P. wing turbines each developing equal power to the centre H.P. turbine with about half the weight of steam at a very much lower pressure.

The B.T.U. or heat drop for a given pressure drop *increases with the fall of pressure*, and as the kinetic energy given up to the blades entirely depends on the *heat drop*, it naturally follows that in the case of very low-pressure steam the same amount of work can be obtained with a much smaller pressure drop, as also shown in the example from actual practice given on page 142. It should be carefully noted that in turbine practice the kinetic energy got out of

the steam at each stage or pair of rows varies directly as the drop in heat units, and is, strange as it may appear, quite independent of the pressure carried. It will also be obvious that a high vacuum will increase the efficiency of the low-pressure turbines by allowing of a further drop in pressure, thus developing to the full extent the benefit of the increasing drop in heat units per given pressure drop. Allowing, therefore, equal powers to be developed in each shaft and in each turbine, it is of interest to note that!—

1. The drop of pressure is less throughout each L.P. turbine to develop the same power as in the H.P. turbine.

2. The amount of steam used in each L.P. turbine is roughly only half that used in the H.P. turbine to develop the same power, as the exhaust from the H.P. turbines divides into two portions, one for each L.P. turbine.

Work done by Difference of Vacuum in L.P. Turbines.—When running at reduced speed with all turbines working (as will be noticed in the example given on page 170), the H.P. turbine carried an initial pressure of 80 lbs. by gauge, and the L.P. turbines each an initial pressure corresponding to 15 in. vacuum, which, worked out, gives an absolute pressure of 7½ lbs., as 15 in. ÷ 2 = 7.5 lbs. vacuum, and 15 lbs. (atmospheric pressure), less 7.5 lbs., is equal to 7.5 lbs. absolute initial pressure. From this it will be apparent that each L.P. turbine, working between an initial vacuum of 15 in. and exhaust vacuum 28 in., develops the *same power* as the H.P. turbine, with an initial pressure of 80 gauge and an exhaust pressure of 7.5 lbs. absolute.

So that,

H.P. turbine	initial pressure	=	80 + 15 = 95 lbs.	absolute pressure.		
„ „	exhaust	„	=	(say) 8½ lbs.	„	„
L.P. turbines	initial	„	=	7.5 lbs.	„	„
„ „	exhaust	„	=	(say) 2 lbs.	„	„

NOTE.—The actual back pressure against the H.P. turbines will be about one pound or so in excess of the L.P. initial pressure, and the back pressure against the L.P. turbines about a pound in excess of the condenser pressures.

Referring to the above figures, it will be evident that each L.P. turbine, working with a total pressure drop of 7.5 − 2 = 5.5 lbs. only, develops or gives out fully the same total heat drop, and therefore the same developed power as the H.P. turbine, working at a total pressure drop of 95 − 8.5 = 86.5 lbs. This fact brings out, in a very striking manner, the high heat value of very low-pressure steam when applied to steam turbine practice, and affords perhaps the greatest contrast of all to reciprocating engine practice.

Pressure Drop in H.P. and L.P. Turbines.—The pressure drop per row is less in the L.P. turbines than in the H.P., as will be seen by referring to the practical example given on page 142, and the great difference in volume of one pound of steam at high pressure

and one pound of steam at low pressure is shown by the following figures :—

H.P. Turbine.

At 158 lbs. pressure absolute the volume is 2.81 cub. ft.
„ 160 lbs. „ „ „ 2.78 cub. ft.

The difference = .03 cub. ft. { for a pressure drop of 2 lbs.

L.P. Turbines.

At 10 lbs. pressure absolute the volume is 37.84 cub. ft.
„ 10.5 lbs. „ „ „ 36.14 cub. ft.

The difference = 1.70 cub. ft. { for a pressure drop of .5 lb.

The above demonstrates the necessity for longer blades and wider spacings at the exhaust ends of the L.P. turbines.

Drop of Pressure.—Between each pair of rows of blades the steam drops a certain amount in pressure, and the approximate drop can be shown as follows :—Suppose the initial pressure in the H.P. turbine to be 140 lbs. gauge, and the terminal pressure at the last row of blades to be, say, 22 lbs. gauge, then,

$140 - 22 = 118$ lbs. total drop of pressure in H.P. turbine.

If, then, the H.P. turbine is made up of, say, 60 rows of blades in all, and we divide the total drop by the total number of rows, we obtain the average drop at each row, thus—

118 lbs. $\div 60$ rows $= 1.96$ lb. average drop per row.

NOTE.—It should be noted that there will also be 60 rows of fixed or casing blades.

Again, take the L.P. turbine, also formed of 60 rows of blades, the initial pressure being, say, 20 lbs. gauge, and the condenser back pressure $1\frac{1}{2}$ lbs. (27 in. vacuum), or, say, $2\frac{1}{2}$ lbs. absolute actual pressure at last row of L.P. blades.

Then, $20 + 15 = 35$ lbs. absolute initial pressure,
and $35 - 2.5 = 32.5$ total drop in pressure in L.P. turbine;
then, $32.5 \div 60 = .54$ of a lb. average drop per row.

It should be noted that this drop takes place simultaneously in each L.P. turbine, as the H.P. exhaust divides into two branches, one for each L.P. turbine. The pressure drop gradually *decreases* from the initial end to the exhaust end of each turbine.

Increasing Steam Velocity.—The velocity acquired by the steam at any given position of the rotor entirely depends on the drop produced by the work done or loss of energy at each row, and as the steam expands in falling in pressure it follows that the area allowed for steam flow must be increased to give equal velocities throughout the turbine. As, however, a Parsons type turbine is composed of a

certain number of rows of blades, all of the *same size*, for each expansion, the velocity of the steam at the after rows will be more than that at the forward rows.

To take a practical example. A certain Parsons marine turbine consists, say, of the following :—

H.P. turbine	= 4 expansions,	each	containing	16 rows	= 64 rows.
L.P. "	= 8 "	"	"	8 "	= 64 "
L.P. "	= 8 "	"	"	8 "	= 64 "

Total number of rows of blades, 192

Therefore, if the steam velocity at the first row of the H.P. turbine is, say, 250 feet per second, at the last or sixteenth row of that expansion the velocity will be more, as the volume, due to pressure drop, is increased, with the same blade heights and openings. This holds good for each of the various expansion sets or rows, and the increase of steam velocity can be approximately determined by comparing the volume of steam per pound for each of the two pressures, initial at first row, and terminal at last row.

The difference in blade heights is more marked in the L.P. turbines, as the volume increases very rapidly with fall of pressure in the case of low-pressure steam. The following figures will perhaps make this clear :—

Steam Pressures and Volumes.

Pressure (Absolute).	Cubic feet of Steam per pound weight (Steam volume).	Pressure (Absolute).	Cubic feet of Steam per pound weight (Steam volume).
210 lbs.	2.15 cubic feet.	55 lbs.	7.61 cubic feet.
195 "	2.31 "	35 "	11.64 "
175 "	2.55 "	15 "	25.84 "
155 "	2.87 "	10 "	37.84 "
135 "	3.26 "	8 "	46.68 "
115 "	3.80 "	6 "	61.20 "
95 "	4.54 "	4 "	89.63 "
75 "	5.68 "	2 "	172.08 "

Notice that one pound of steam at 210 lbs. absolute pressure occupies a volume of 2.15 cubic feet, and one pound at 2 lbs. absolute pressure a volume of 172.08 cubic feet.

Pressure Drop, Number of Rows, and Revolutions.—For a given diameter of rotor and required turbine efficiency—

1. A small pressure drop per row produces a low steam velocity, demanding a corresponding low revolution speed, and requiring a large number of rows to absorb the available heat energy of the steam.

2. A large pressure drop per row produces a high steam velocity, demanding a corresponding high revolution speed, and requiring a smaller number of rows to absorb the available heat energy of the steam.

NOTE.—In No. 1 case the revolution speed may be reduced if the diameter of the rotor be increased (as described on page 32), as this will also give the required high peripheral blade speed necessary to still maintain the ratio of $\frac{V_t}{V_s}$ and give constant turbine efficiency.

From 8 lbs. pressure (absolute) down to 2 lbs. pressure (absolute) the increase of steam volume per pound is very marked, and complicates the design of the L.P. turbines, owing to the necessity for allowing suitable blade heights and openings to prevent rapid increase of steam velocity at the low pressures referred to. The diagram of Blade Heights and Steam Volumes compared, facing page 26, illustrates this point to some degree.

Number of Blades per Row.—Regarding the number of blades per row it is of importance to note that in the H.P. turbine and reverse turbines the total number of blades in each row of the rotor usually exceeds the total number in each corresponding row of the casing, and the arrangement is carried out throughout the turbine or cylinder, whereas, in the L.P. turbines, the number of rotor blades in each row of the first four expansions only generally exceeds the number of blades in the casing rows, as throughout the remaining four expansions the number of rotor blades per row is usually less than those in the casing, this being necessary to allow of the exceptionally large volume of steam produced at the last L.P. expansions to flow through the moving blades without attaining an excessive velocity.

Special tools are now being used either to "close up" or "open out" the exit edges after the blades are already fixed in position, as it is upon the correct adjustment of the blade exit area that the efficiency and economy of the turbines greatly depend, and evidently the only true guide to which is actual experiment, theoretical calculations evidently being in many cases unreliable and misleading, as it is quite a common experience for the blade openings to be increased after the turbines are bladed.

Tip Leakage and Dryness Factor.—Owing to blade tip leakage, the adiabatic expansion of the steam may be said to be incomplete, as the leaking steam partly superheats the expanded and lower pressure steam further on, thus raising the dryness factor above the calculated amount obtained by pure adiabatic expansion alone, and proportionally reducing the heat drop and work done. A reference to the "error" diagram and "Table of Actual Volumes," page 41, together with the Rule for work done by adiabatic expansion on page 34, will perhaps make this clear.

Turbine Efficiency.—The principal turbine losses are usually estimated as follows :—

Friction of steam against blades, } 20 per cent.
" " " itself, }
Leakage over blade tips, 8 per cent.
" " dummy piston at initial end of turbines, }
" " steam glands at both ends of turbines, } 5 per cent.
Exhaust steam heat loss, }
Mechanical friction of bearings, &c., 8 per cent.

These losses in efficiency have been found to total up to somewhere about 40 per cent., thus leaving an actual mean efficiency of 60 to 62 per cent.

To express the efficiency another way.

Steam consumption in theory ÷ steam consumption in practice = .60 to .62.

From various cases taken from actual practice the above figures work out as the average efficiency. The steam and blade friction losses are lowest at the H.P. turbine, and highest at the L.P. turbines, owing to the increase of friction at the longest blade heights.

Actual Volume of Steam.—According to the "error" diagram of steam volumes supplied by Mr E. M. Speakman (see page 21), the actual steam volume or dryness value in the turbine is less than the specific volume by from about 4 per cent. at 80 lbs. absolute pressure to 20 per cent. at 2 lbs. absolute pressure, the volume of steam in the turbine occupying to a certain degree an intermediate position between the specific volume and the volume due to adiabatic expansion. The actual steam volume can therefore be determined as follows :—

$$\text{Actual Steam Volume} = \frac{\text{Specific Volume} \times 100}{100 + \text{per cent difference}}.$$

EXAMPLE.—Determine the actual steam volume per pound in an L.P. turbine if the pressure is 15 lbs. absolute, and the error difference 11 per cent. (by diagram, page 21).

Then, 15 lbs. absolute = 25.84 cub. ft. Specific Volume (from Steam Table),

$$\text{and, } \frac{25.84 \times 100}{100 + 11} = 23.27 \text{ cubic feet Actual Volume.}$$

NOTE.—It will be seen by referring to the error diagram that the actual steam volume at the above pressure is also fully 4 per cent. *more* than that due to adiabatic expansion.

Actual Steam Volumes in Turbine.—The following table of actual steam volumes per pound and dryness factors to correspond is calculated from the error diagram given by Mr E. M. Speakman in his paper on "The Steam Turbine with Special Reference to Marine Work," and gives a fair idea as to the required relative proportions between blade heights, blade angles, and steam speeds at

the various pressures existing in the turbine casings at the different expansions.

Table of Steam Volumes.

Absolute Pressure in lbs.	Specific Volume in Cubic Feet per lb.	Actual Volume in Turbine in Cubic Feet per lb.	Dryness Factor.
100	4.33	4.18	.965
80	5.34	5.12	.958
60	7.01	6.66	.950
55	7.61	7.19	.945
50	8.32	7.84	.943
45	9.19	8.64	.940
40	10.26	9.61	.936
35	11.64	10.85	.932
30	13.46	12.46	.925
25	15.97	14.65	.917
20	19.71	17.91	.908
15	25.84	23.23	.898
10	37.84	33.48	.884
9	41.77	36.80	.881
8	46.68	40.94	.877
7	52.93	46.22	.873
6	61.20	53.10	.867
5	72.99	62.92	.862
4	89.63	76.60	.854
3	117.50	99.66	.848
2	172.08	143.70	.835
1.5	225.58	185.66	.823

Referring to the error diagram mentioned (page 21), it will be seen that at, say, 10 lbs. pressure absolute, the actual steam volume in the turbine is less than the specific volume of dry steam by about 12½ per cent., and more than the volume due to adiabatic expansion by fully 5 per cent.

Velocity Triangles and Horse-Power.—The work done per pound of steam flow and the horse-power developed depends on the initial velocity and exit velocity of the steam, together with the efficiency of the blades.

The steam exit velocity and mean blade velocity being known, the initial velocity of the steam can be determined by the geometrical construction of a velocity triangle, given, of course, the blade angle of the "expansion" under consideration.

The exit angle given to the blades largely determines the exit openings available for steam flow, and therefore governs the exit

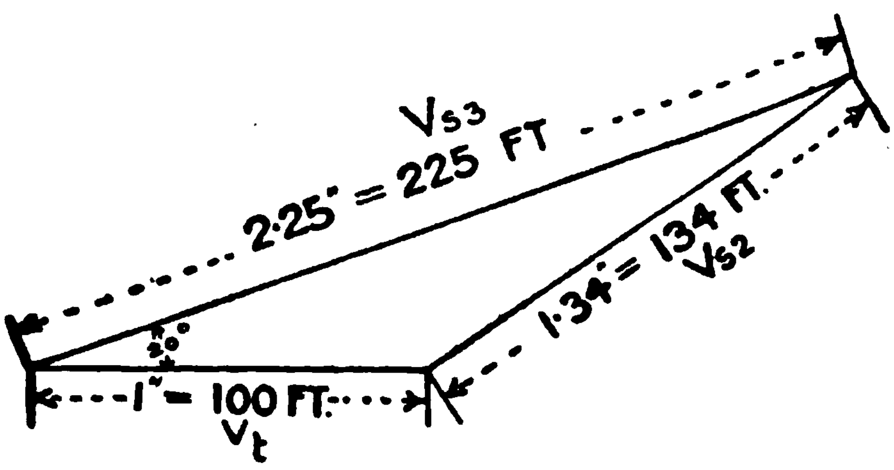
steam velocity; this, again, together with the blade velocity, regulates the admission velocity of the steam, and consequently the work done per blade row. The actual areas of exit openings can be accurately determined by mathematical calculations, or by actual measurement, after which the steam exit velocity can be found if the weight of steam flow and actual volumes of the mixture (steam and water) are known. From these, a "velocity triangle" can be constructed, and the theoretical velocity of steam admission determined.

EXAMPLE 1.—Construct a velocity triangle, and determine the horse-power developed at the first H.P. expansion of 16 rows, given the following:—

Mean diameter across blades - - -	48 inches.
Revolutions - - - - - - - -	480 per minute.
Steam exit velocity - - - - -	225 feet per second.
Steam flow per minute - - - -	2,400 lbs.
Assume blade efficiency as - - -	85 per cent.
Scale of triangle - - - - -	100 feet per inch.
Blade angle - - - - - - -	20° (normal blades).

$$\text{Then, } V_t \text{ (blade velocity)} = \frac{48 \times 3.1416 \times 480}{12 \times 60} = 100 \text{ feet per second.}$$

Set off the angle of 20° as shown, and measure up by the scale the steam exit velocity of 225 feet, which will be equal to 2.25 inches, then set off horizontally the blade velocity of 100 feet, equal to 1 inch; now complete the triangle, and the third side, which will be found to measure 134 feet (or 1.34 inches), represents the initial velocity of the steam.



Velocity Triangle H.P. Turbine.

The work done in Foot-pounds per minute in this expansion =

$$\frac{(225^2 - 134^2) \times 2400 \times 16 \text{ rows} \times .85}{64.4} = 16557704 \text{ foot-pounds.}$$

$$\text{Horse-power} = 16557704 \div 33000 = 501.7.$$

EXAMPLE 2.—Repeat the foregoing for the first L.P. expansion of 8 rows, given the following:—

Mean diameter across blades - - -	68 inches.
Revolutions - - - - - - -	480 per minute.
Steam flow - - - - - - -	1,200 lbs. per minute.
Actual steam volume - - - -	12.04 cubic feet per lb
Blade angle - - - - - - -	20° (normal blades).
Assume blade efficiency as - - -	74 per cent.
Blade heights - - - - - -	$1\frac{1}{2}$ inches.

$$\text{Then, } V_t = \frac{68 \times 3.1416 \times 480}{12 \times 60} = 142 \text{ feet per second.}$$

$$V_s = \frac{\text{cubic feet steam flow per second}}{\text{clear area through blades}} = \text{feet per second.}$$

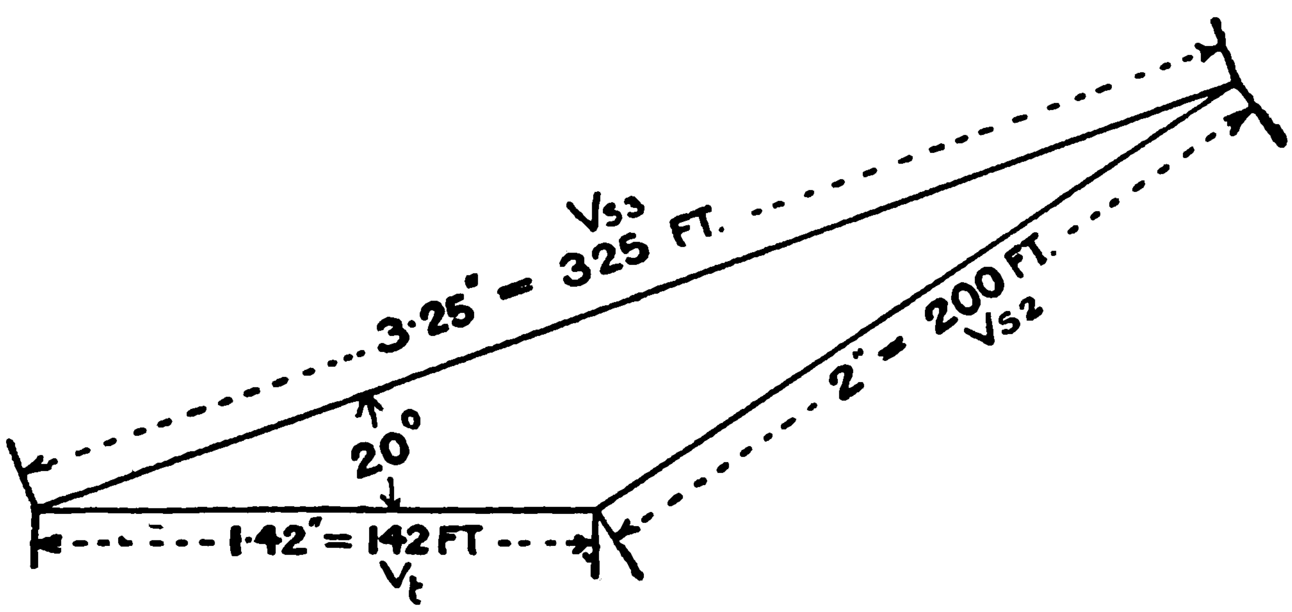
$$\text{Cubic feet flow per second} = \frac{1200 \times 12.04}{60} = 240.8 \text{ cubic feet.}$$

$$\text{Clear area between blades} = \frac{68 \times 3.1416 \times 1.5}{3 \times 144} = .741 \text{ square foot.}$$

Therefore, $V_s = 240.8 \div .741 = 325$ (nearly) feet per second.

NOTE.—For normal blades divide by 3 for clear area, and by 144 to obtain square feet.

$$\text{Then, } V_t = \frac{68 \times 3.1416 \times 480}{12 \times 60} = 142 \text{ feet per second.}$$



Velocity Triangle L.P. Turbine.

Set off the angle of 20° and measure up the exit steam velocity of 325 (3.25 inches) as shown, then set off 142 feet (1.42 inches) as the blade speed, and by completing the triangle the initial velocity of the steam measures 200 (2 inches nearly) feet per second.

$$\text{Work done at this expansion} = \frac{(325^2 - 200^2) \times 1200 \times 8 \times .74}{64.4}$$

$$= 7239130.43 \text{ foot-pounds.}$$

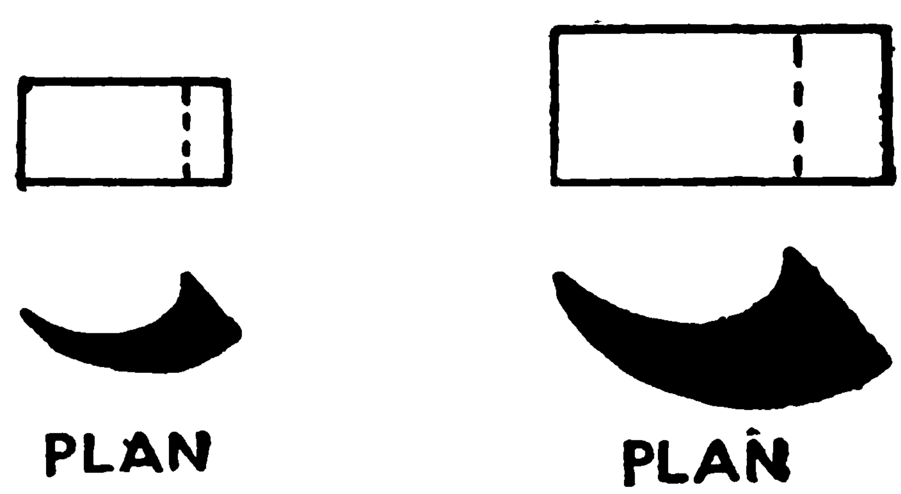
$$\text{Horse-power} = 7239130.43 \div 33000 = 219.3.$$

Number of Expansions.—The usual number arranged is 4 expansions in the H.P. turbine, and 8 expansions in each L.P. turbine, the total number of rows of blades being the same for each separate turbine. So that if the H.P. turbine is made up of, say, 4 expansions, each containing 16 rows of blades, then each L.P. turbine will have 8 expansions, each containing 8 rows of blades. Or again, suppose the H.P. turbine to consist of 4 expansions with 14 rows of blades in each, then the L.P. turbines will consist of 8 expansions with 7 rows of blades in each. It must of course be understood that the rotors will in this case contain 56 rows in all, and the casings 56

rows in all; or 56 rows of moving blades and 56 rows of fixed blades for each turbine, H.P. and L.P. A pair of rows, consisting of one row of fixed and one row of moving blades, is usually called a "stage." So that in the above case the H.P. turbine will have 14 stages in each expansion, and the L.P. turbines have 7 stages in each expansion.

The "packing pieces," which are caulked in between the blades, vary in shape according to position, being of small thickness section and much curved at the first expansions, and of larger thickness section and less curvature at the last expansions. The shape of the packing piece exactly corresponds to the two surfaces of the blades between which it fits in. So that there is a much smaller curve on the side in contact with the steam-receiving surface of the blade than that in contact with the convex surface or back of the blade.

Packing Pieces.—The small packing pieces which are caulked in between every pair of vanes are made of brass, and vary in size according to the position occupied by them in the turbine, being small



"**Packing Pieces**" **(full size).**

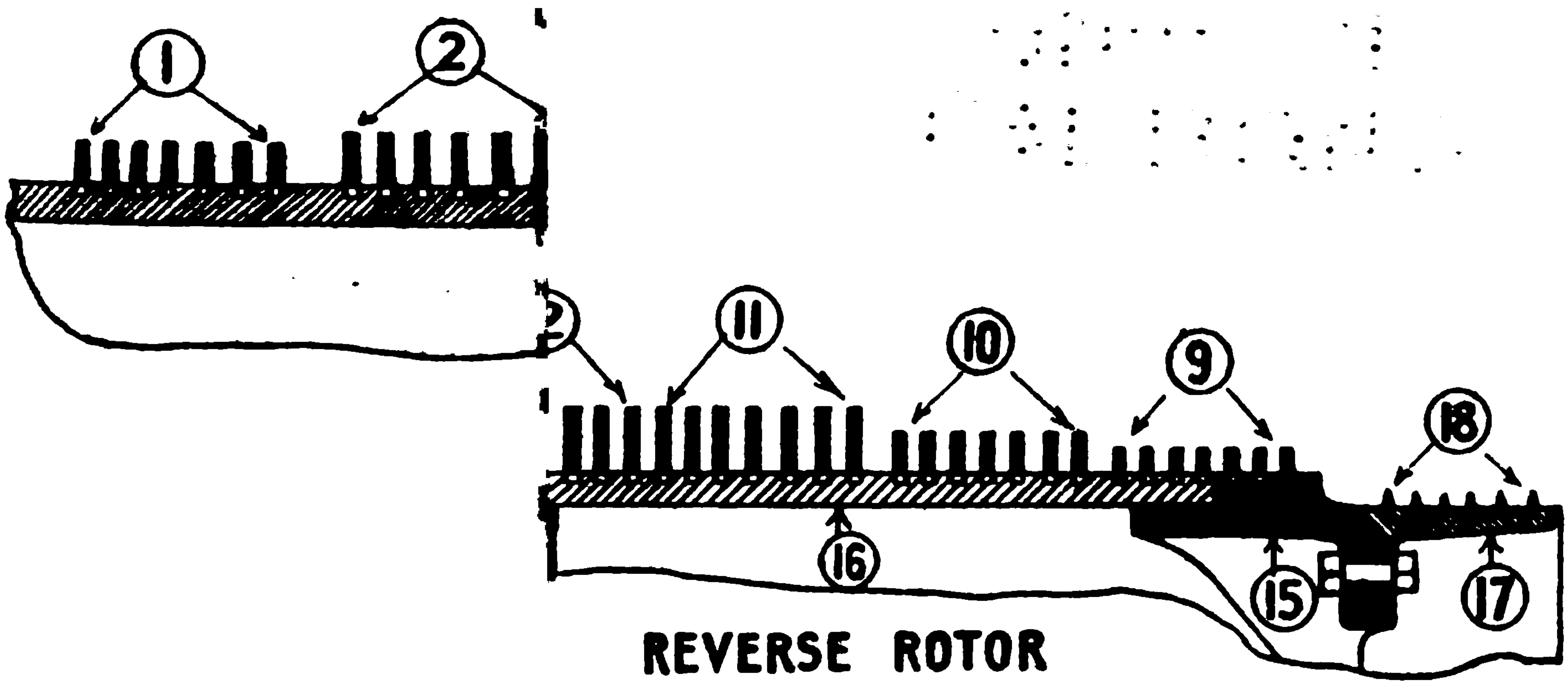
at the forward or higher pressure end, and larger at the after or lower pressure end (see sketch).

Effect of opening up of blades:—

1. With a limited steam flow per second the velocity of the steam will be reduced with loss of efficiency.

2. With unlimited steam supply the velocity may remain constant, but the quantity of steam flowing per second will then increase, thus developing more speed and power in the turbines.

In a case which recently came under the writer's notice, the turbines of a steamer did not develop the full power during trials;



hts.

In this turbine four H.P. expansions. The reverse rotor contains

(*Note*— not the same number of blades per row.)

peed 20 knots.

fth Expansion. (6) Sixth Expansion.

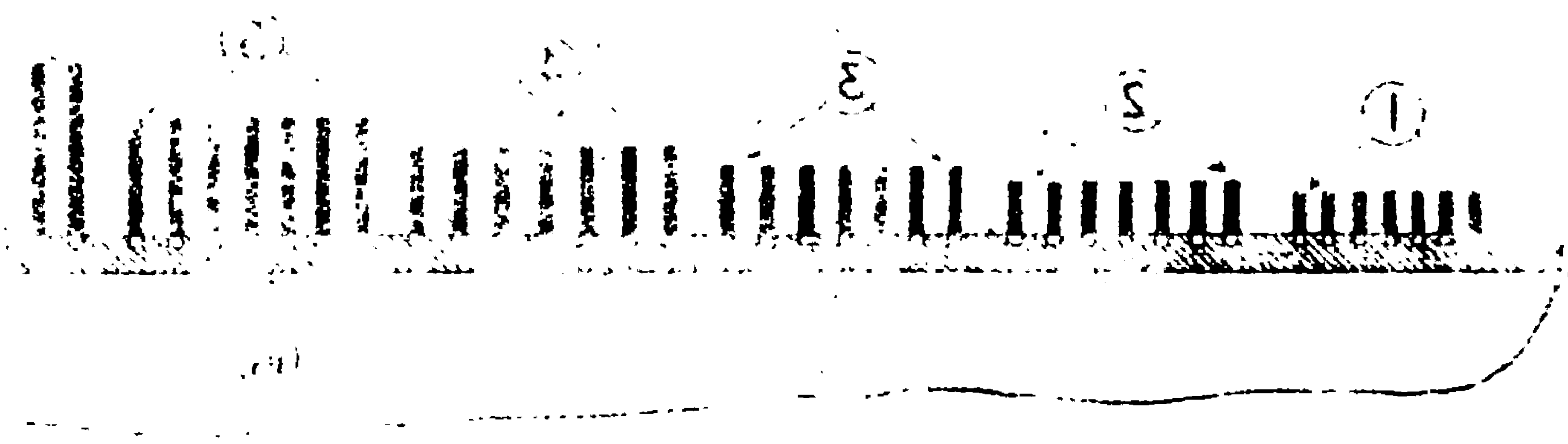
ides).

Fourth Expansion.

(piston. (18) Reverse dummy "fins."

Fourth Expansion.

[*To face page* 44.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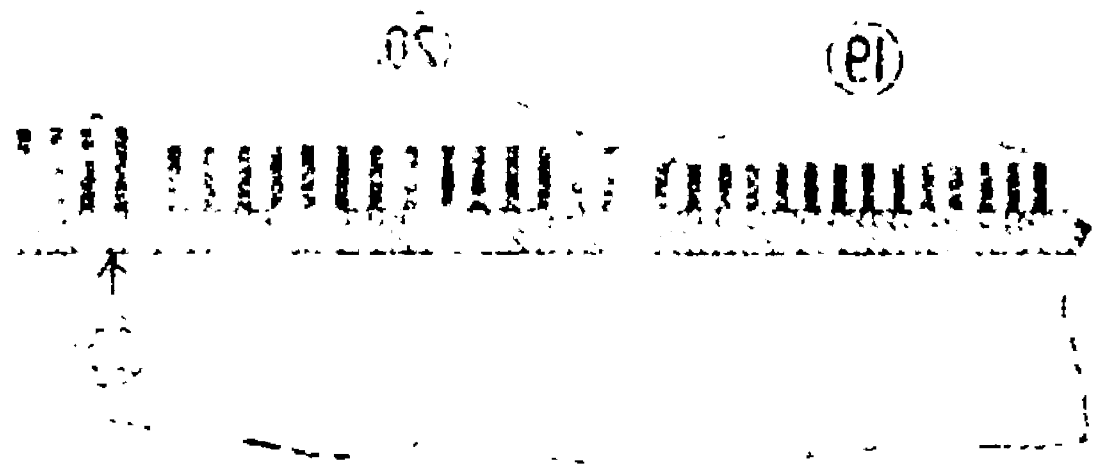
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as a remedy the turbine covers were lifted, and a staff of men set to work to ***open up*** the exit edges of the blades of the first two or four rows in each L.P. expansion; this alteration had the desired effect of inducing a quicker steam flow per second, and in increasing the power of the turbines and the speed of the steamer to that required by the contract. Before the alteration the turbines did not take away the steam fast enough from the boilers, notwithstanding the fact that the stop-valves were full open. This occurrence is unusual. Before the alteration the L.P. turbine receivers indicated a high pressure, but afterwards this pressure fell off to a normal figure. Obviously the L.P. turbines were unable to use up the steam, thus producing a high back pressure against the H.P. turbines, and resulting in reduced power in all the turbines.

"Wing" Blades.—At the 6th, 7th, and 8th L.P. expansions the blade heights are usually the same, but the 7th and 8th expansions are fitted with blades of much flatter section than the normal blades, and which give a much increased area of exit opening; these blades are termed "wing" blades, and are made in three types, "semi-wing," "wing," and "double-wing." The usual arrangement is to fit "semi-wing" and "wing" blades in the 7th and 8th L.P. expansions, but in the large Cunarders "double-wing" blades were also fitted.

The "wing" type of blades give an area ranging from about 50 per cent. to 85 per cent. of the annulus area, which allows of moderate blade heights at the last L.P. expansions where the steam volumes to be handled may be anything from 40 to 100 cub. ft. per pound.

"Wing" blades are also fitted in the last reverse turbine expansions.

In blading lists as issued to the blading staff, the various blades are indicated by a three-figure number, the second figure of which represents the number of eighths in the blade widths, and the last figure the type of blade, whether "normal," "semi-wing," "wing," or "double-wing."

Letter.	Last Figure.
B = blades.	0 = "normal."
C = casing.	1 = "semi-wing."
S = rotor.	2 = "wing."
	3 = "double-wing."

Thus,	230 means	"normal"	$\frac{3}{8}$-in.	blades.
	240	,,	,,	$\frac{1}{2}$-in.	,,
	251	,,	"semi-wing"	$\frac{5}{8}$-in.	,,
	262	,,	"wing"	$\frac{3}{4}$-in.	,,
	263	,,	"double-wing"	,,	,,
	302	,,	"wing"	$1\frac{1}{4}$-in.	,,
	381	,,	"semi-wing"	1-in.	,,

Therefore, 130 B means blades $\frac{3}{8}$ in. wide and "normal."
130 C ,, casing packing pieces $\frac{3}{8}$ in. wide and "normal."
130 S ,, rotor ,, ,, ,, ,,
240 B ,, blades $\frac{1}{2}$ in. wide and "normal."
240 C ,, casing packing pieces $\frac{1}{2}$ in. wide and "normal."
240 S ,, rotor ,, ,, ,, ,,
251 B ,, blades $\frac{5}{8}$ in. wide and "semi-wing."
251 C ,, casing packing pieces $\frac{5}{8}$ in. wide and "semi-wing."
251 S ,, rotor ,, ,, ,, ,,
262 B ,, blades $\frac{3}{4}$ in. wide and "wing."
262 C ,, casing packing pieces $\frac{3}{4}$ in. wide and "wing."
262 S ,, rotor ,, ,, ,, ,,

The following blading table gives the usual dimensions of blades casing packing pieces, and rotor packing pieces, and it will be observed that for heavy blading the rotor grooves are deeper than those of the casing.

No. and Letter (Blades).	No. and Letter (Casing Packers).	Depth of Groove.	Width of Groove.	No. and Letter (Rotor Packers).	Depth of Groove.	Width of Groove.
120 B	120 C	$\frac{1}{4}$ in.	.243 in.	120 S	$\frac{1}{4}$ in.	.243 in.
130 B	131 C	$\frac{1}{4}$ „	.415 „	131 S	$\frac{1}{4}$ „	$\frac{13}{32}$ „
240 B	240 C	$\frac{3}{8}$ „	.510 „	240 S	$\frac{3}{8}$ „	.510 „
252 B	252 C	$\frac{3}{8}$ „	.716 „	252 S	$\frac{1}{2}$ „	.716 „

NOTE.—Middle figure of blade No. = width in eighths of an inch.

Last „ „ = "normal" (0), "semi-wing" (1), "wing" (2), or "double-wing" (3).

Blade Speed and Steam Speed Ratio.—The ratio V_t to V_s at any expansion can be determined as follows:—

EXAMPLE.—Compare the ratio of V_t to V_s at the 1st H.P. expansion and at the 8th L.P. expansion, the steam flow being 31 lbs. per second through the H.P. turbine, and 15.5 lbs. per second in the L.P. turbine.

The revolutions are 470 per minute, the H.P. rotor drum 48 in. diameter, and the L.P. rotor drum 68 in. diameter, the H.P. 1st expansion blades $1\frac{1}{2}$ in. in height, and the L.P. 8th expansion blades 8 in. in height.

The actual steam volume per pound at the H.P. 1st expansion is 2·88 cub. ft., and at the 8th L.P. expansion 140 cub. ft.; the clear exit area for steam flow is one quarter of the H.P. annulus area, and one half of the L.P. annulus area.

H.P. Turbine.

$$\text{Then, } V_t = \frac{\text{Mean blade circumference} \times \text{revolutions}}{60} =$$

$$\frac{(48 \text{ in.} + 1\tfrac{1}{2} \text{ in.}) \times 3.1416 \times 470}{12 \times 60} = 101.5 \text{ ft. per second.}$$

NOTE.—12 in. to reduce to feet.
NOTE.—60 seconds to reduce to seconds.

$$\text{Exit area} = \frac{\text{Mean blade circumference} \times \text{blade height}}{\text{clear area ratio}} =$$

$$\frac{(48 + 1.5) \times 3.1416 \times 1.5}{4 \times 144} = .404 \text{ sq. ft.}$$

NOTE.—144 sq. in. to reduce to sq. ft.

$$\text{Therefore, } V_s = \frac{\text{lbs. steam per second} \times \text{volume}}{\text{exit area}} = \frac{31 \times 2.88}{.404} = 220 \text{ ft. per sec.}$$

$$\text{Then, Ratio } V_t \text{ to } V_s = 101.5 \div 220 = .46.$$

L.P. Turbine.

$$\text{Again, } V_t = \frac{(68 + 8) \times 3.1416 \times 470}{12 \times 60} = 155 \text{ ft. per second.}$$

$$\text{Exit area} = \frac{(68 + 8) \times 3.1416 \times 8}{2 \times 144} = 6.63 \text{ sq. ft.}$$

$$\text{Therefore, } V_s = \frac{15.5 \times 140}{6.63} = 327.$$

$$\text{Then, Ratio } V_t \text{ to } V_s = 155 \div 327 = .47.$$

Cruising Turbines.—In Admiralty work cruising turbines are fitted to allow of low speeds and powers, together with fair economy. The cruising turbines are distinctive from the ordinary turbines in the following points:—

1. Longer rotor drums, owing to the greater number of blade rows required per expansion.

2. Smaller blade differences per expansion, this being necessary to keep down the difference of pressure and steam velocity, so that with the reduced mean blade speed required, the ratio of blade speed to steam speed will not be far from .43 to .48.

3. Smaller blade heights and rotor diameter due to smaller steam flow per second for the reduced H.P.

The number of expansions are usually either 3 or 4, and each expansion consists of from 16 rows to 22 rows of blades. In cruisers of the "Bellerophon"-"Boadicea" class the cruising turbines are fitted with blade heights as follows:—

1st expansion, 22 rows of blades,	$\frac{9}{16}$ in.	in height, at	1 in.	pitch.
2nd " " " "	$\frac{11}{16}$ in.	"	1 in.	"
3rd " " " "	$\frac{15}{16}$ in.	"	$1\frac{1}{4}$ in.	"
4th " " " "	$1\frac{1}{4}$ in.	"	$1\frac{1}{4}$ in.	"

The drum diameter being 68 in. and the length 8 ft. 5 in.

(1)
(2) le.
(3) ings.
(4) msbottom rings.
ffle plate.

(13) Main bearing.
(14) Thrust.
(15) Sliding coupling.
(16) Main turbine shaft.

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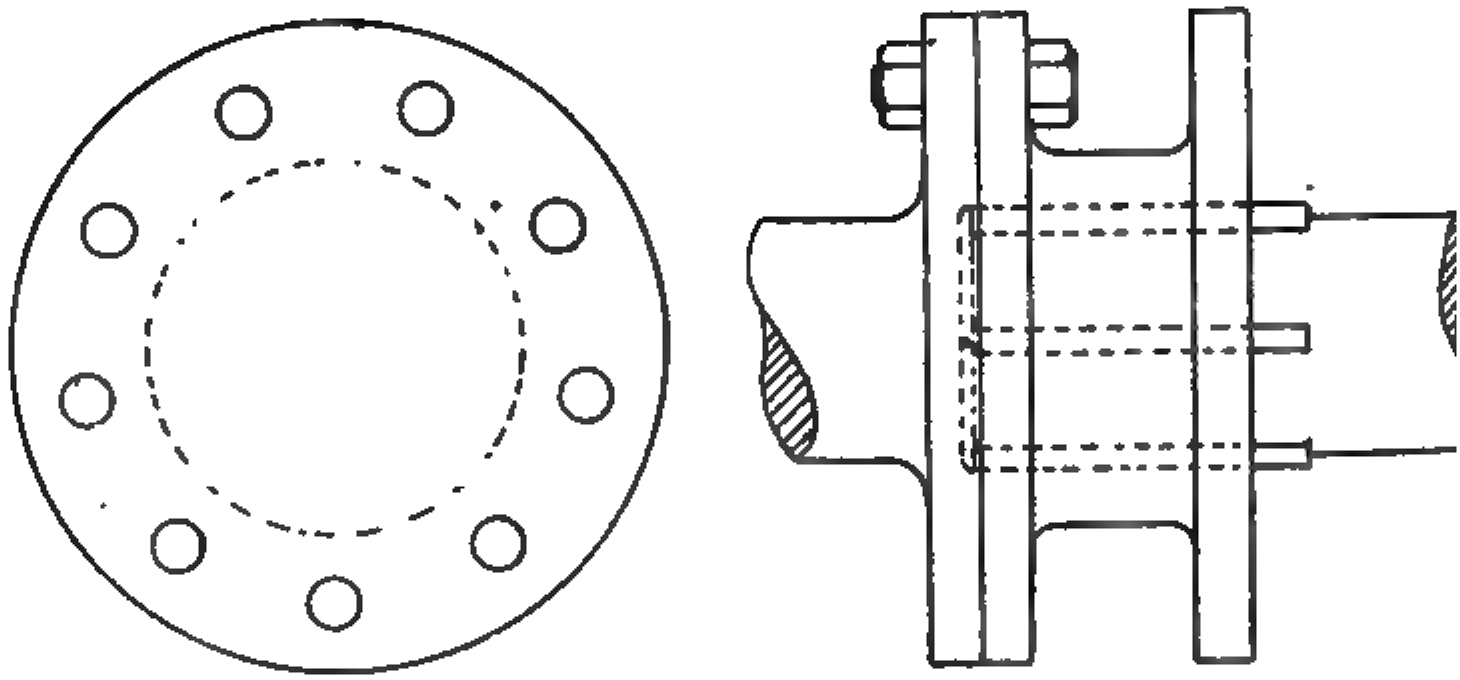
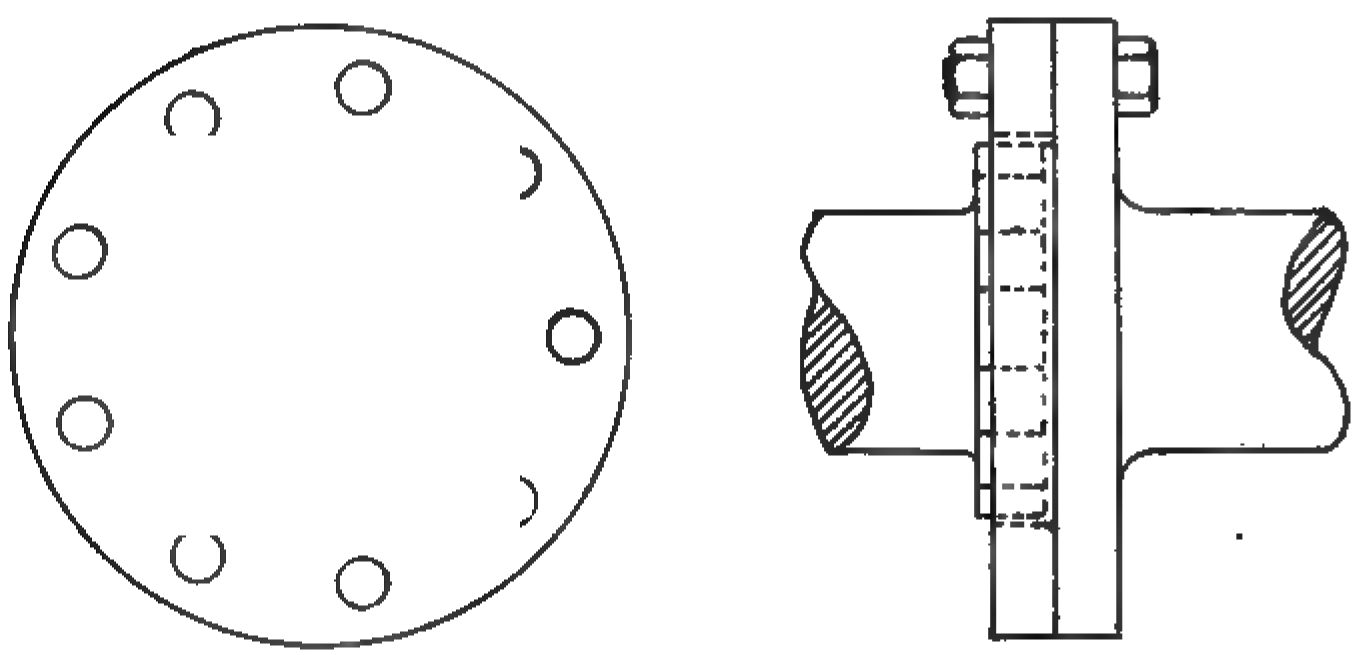


Sliding Coupling with Loose Flange.



Sliding Coupling, with Teeth.

The blade tip clearance (cold) for the above varies from $\frac{40}{1000}$ in. at the 1st expansion to $\frac{60}{1000}$ in. at the 4th or last expansion; when heated up, however, the actual blade tip clearance is only about two-thirds of the foregoing at the 1st expansion, and rather less at the last expansion, the brass blades expanding more than the steel rotor drum or the cast-iron casing. The cruising dummy clearance cold is only about $\frac{15}{1000}$ in., but generally, when heated up, this increases to about $\frac{25}{1000}$ in., or even more. The steam, after passing through the first cruising turbine, enters the second or M.P. cruising turbine, if one is fitted, then the H.P. ahead and L.P. ahead turbines, exhausting finally to the condenser. If only one cruising turbine is fitted to each set, the steam exhausts from it direct to the ahead H.P. turbine, which is the arrangement in the "Indomitable"-"Inflexible" class.

During the trials of the "Indomitable" and "Inflexible" speeds of over 20 knots were easily obtained with the cruising turbines connected up in series with the main turbines.

Adjustment and Expansion Allowance.—The cruising turbines being in line with the main turbines require some type of adjustable coupling for connecting the shafts of each, and the two systems which have been adopted in practice are:—

1. Bolted coupling, with loose or sliding flange on one of the shafts, moving longitudinally on feathers.

2. Flanges arranged to fit together by a system of teeth and recesses arranged circumferentially as shown in the sketch. This method is certainly the simplest, and is used in torpedo boat practice, and in vessels of the "Boadicea" class, although perhaps not so mechanically perfect as the other.

The sliding coupling is necessary to allow of separate "face" dummy clearance adjustment in the cruising turbine and in the ahead turbine, also to provide for independent expansion of each without one affecting the other. In large vessels the cruising turbines are generally fitted with the "face" type of dummy ring similar to the ahead turbines, whereas the astern turbines, whether H.P. and independent, (see "Indomitable"), or incorporated with the L.P., are fitted with the "fin" type of ring. The "fin" type of dummy gives large longitudinal clearance spaces without danger of contact, and thus allows of ample variation of adjustment of the ahead "face" ring dummy without risk of the astern dummy being damaged. This being the case, when H.P. astern turbines are fitted on the H.P. shaft, and independent of the H.P. turbine, a solid coupling is sufficient for connecting up the two turbines (as shown in the sketch), instead of a sliding coupling as required for connecting the cruising turbines.

Sometimes, however, in the case of smaller cruisers or battleships, a solid coupling is employed between the cruising and ahead turbines, and the cruising turbine dummies are then made of the "fin" type, similar to the astern, to allow of adjustment of the ahead turbine on the same shaft, and also for expansion. In this case the cruising

thrust-block receives a share of the propeller thrust, and the steam pressure thrust of the cruising blades acts as a counterbalance to th propeller thrust pressure.

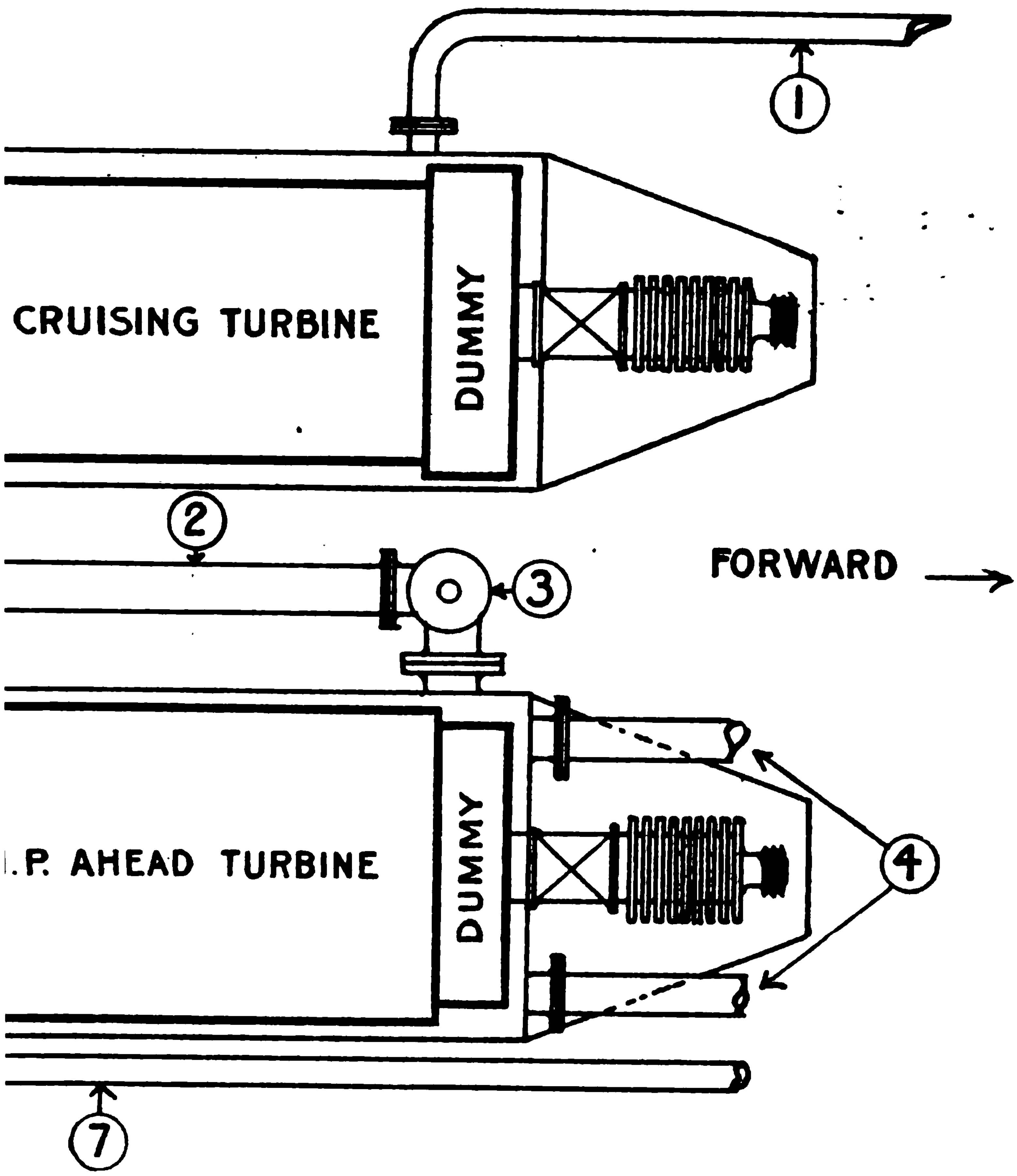
The "fin" tip clearance is usually $\frac{20}{1000}$ or .020.

Cutting Out of Cruising Turbines.—The engineering staff of Government vessel recently made the interesting discovery that t consumption per horse-power at cruising speeds was much about t same when running with the main turbines only and the stop valv shut in as when running with the cruising turbines connected up a receiving boiler steam direct. This no doubt accounts for the rec decision of the Admiralty to, in future, discontinue the fitting cruising turbines to battleships and cruisers. Again, experience proved that the "fan resistance" of the cruising turbine blades wh running ahead with the main turbines only resulted in a somew serious loss of power, and to partly overcome this, it was fou advisable to ensure a vacuum in the cruising turbines *when c* This was done by increasing the fitting of a special "lantern" ste gland on the steam pipes, in addition to the usual expansion jo When running with the main turbines only, it was found that cruising turbines were drawing in air through the expansion jo of the steam pipes, which, although tight enough when heated under steam, were allowing the admission of air when cold, ow probably to the contraction when cooling.

The air thus admitted increased the resistance opposed to cruising blades when revolving idly, and absorbed a certain amo of power. By the fitting up of the "lantern" type of steam gland the steam pipe outside of the ordinary expansion joint, the air is k out, and a vacuum maintained inside the turbine casings, wh reduced the resistance to be overcome by the moving blades, which proportionally set free a certain amount of horse-power wh could be effectively utilised for purposes of propulsion.

As will easily be imagined, the disconnecting of the cruising turb altogether when running ahead would result in still further impro ment, as all drag or resistance would then be taken off the ah turbines. The same remark holds good as regards the astern turbi which, whether independent or incorporated, act as a hindrance to ah running. In like manner, when running astern, the ahead turb act as a resistance, and represent so much lost power, which, if tes would likely be found to be somewhat considerable. The ab constitutes a rather important drawback to the turbine, as, owing the existing conditions, full economy is impossible of attainment.

Cruising Thrusts.—The cruising turbines are fitted with s thrusts similar to those of the ahead turbines, but it should be no that no propeller thrust is taken up by these, as the sliding coup interposed between the ahead turbine in line with the correspond cruising turbine prevents the cruising thrust block from receiving



rn turbine.
nt rotor adjustment.
ı radial fin rings).

lances the propeller thrust forward.
:es the propeller pull aft when running astern.
: the propeller pull aft when running astern.
the blades aft.(no propeller thrust takes effect on this turbine owing to

anchored aft and free to expand forward by means of oval holes for the
[seating bolts at that end.

[To face page 48.

pressure from the propeller, the ahead turbine taking up its own thrust and that produced by the cruising turbine in addition. The cruising thrust block is required for two reasons :—

1. To allow of the rotor and dummy adjustment.

2. To take up the *steam* thrust on the blades, which is exerted from the initial to the exhaust end.

From the foregoing it will perhaps be obvious that, should the cruising turbines be worked at an overload, the ahead turbine thrust of that shaft may be in danger of overheating owing to the excessive thrust pressure set up, and which may not be balanced by the reduced pressure carried in the ahead turbine when running with the cruising turbine working. The ahead turbine thrust, if designed only for the full pressure carried in that turbine, may not have sufficient surface to take up the combined propeller thrust of its own and the cruising turbine, because the steam thrust on the ahead blades being less and the cruising turbine taking none of its own propeller thrust, nor yet giving any counterbalancing steam thrust aft on the shaft, the dangers of excess pressure will be evident.

Cruising Turbine Glands.—The cruising turbine glands are similar to those of the H.P. turbines, being fitted with a pocket to which steam is led when heating up, and which afterwards acts as a "leak-off" when running.

Cruising Turbine Drains.—The cruising turbine drains are led away to the "after" end of the L.P. turbines, and are thus in direct connection with the wet air pumps. When running, the drain valves at the exhaust end are only kept slightly open, and the drains at the initial end shut. When stopped, however, all the drains are opened up full to clear the turbines of condensed water. If the drains were full open when running, steam would, of course, be blown out with the water.

Regarding the foregoing, the naval practice is as follows :—

Previous to lighting fires.	Open screw-down drain valves at both ends of cruising turbines, ahead ends of L.P. turbines (main), and after end of H.P. main turbines.
With H.P. cruising turbines under steam.	All screw-down drains to be closed down, except drains at exhaust ends of cruising turbines which should be kept slightly eased.
With M.P. cruising turbines under steam (H.P. cruising shut off).	Drains on H.P. cruising turbines to be slightly eased.
With H.P. and M.P. cruising turbines shut off.	Drains on both to be slightly eased.

Steam Connections.—When the cruising turbines are arranged on the compound system boiler steam is led direct to the H.P. cruising, and exhausts from the H.P. to the I.P. cruising (on another shaft),

from there the steam exhausts into the H.P. ahead turbine, and L.P. ahead turbine, finally passing to the condenser. This method of working is arranged for at the lowest powers and speeds; for higher speeds the boiler steam is led direct to the I.P. cruising turbine, then the H.P. ahead, L.P. ahead, and condenser. For full powers and speeds the cruising turbines are cut out altogether, and the boiler steam led direct to the H.P. turbine, then L.P. and condenser.

The cruising turbines are only fitted for ahead running. In the "Indomitable"-"Inflexible" class of vessel with double compound ahead turbines, two cruising turbines are fitted, both being H.P., so that when running at low speeds and powers the boiler steam is led direct to each cruising turbine, and from there to the corresponding H.P. ahead and L.P. ahead turbines and so to the condenser of that set.

Reverse Turbines.—The reverse turbines consist of a set of smaller blades placed in the after end of the L.P. turbine casings. The angle of these blades is the same as the ahead, but the curve is on the other side, and as the steam is admitted at the *opposite* end of the turbine casing, the direction of shaft rotation is reversed. Both the ahead and reverse turbines exhaust into the same central exhaust passage leading to the condenser. The reverse turbines are also fitted with dummy pistons, fitted with "radial fin" type of rings.

The reverse casing is fitted with small drain holes (in the lower half), one hole for each expansion (at the end row of each). The water so drained off passes into the exhaust space, and then into the condenser.

In the latest types of turbine steamers the size of the reverse turbines has been considerably increased.

The reverse rotor is often made of the same diameter as the H.P. rotor, and the reverse power developed equal to about one-half or even more of the ahead power.

H.P. and L.P. Astern Turbines.—In naval practice astern turbines are often fitted to each shaft, and if four lines of shafting, these are arranged as H.P. and L.P. astern, two of each in all, the H.P. asterns being placed on the ahead H.P. turbine shafts, and the L.P. asterns on the ahead L.P. turbine shafts.

In running astern, steam is led first to the H.P. astern turbines from which it exhausts to the L.P. astern turbine, and from there to the condenser as usual.

The H.P. astern turbines are independent of the H.P. ahead turbines, but the L.P. astern turbines are incorporated with the ahead L.P. turbines. The initial end of the H.P. astern is *forward*, and the initial end of the L.P. astern *aft*, which arrangement requires a difference in the dummy diameter of each, the H.P. astern dummy being in excess of the corresponding rotor drum diameter, and the L.P. astern dummy being less than the corresponding rotor drum diameter. This allows of an annulus, on which the steam pressure will, in both H.P. and L.P. asterns, act *forward* and thus balance to a certain degree the propeller pull aft when running astern.

The following show the respective diameters of rotor drum and dummy for ahead, astern, and cruising turbines:—

Turbines.	Drum Diameters.	Dummy Diameters.
H.P. ahead -	90 inches.	90 inches.
H.P. astern -	80 "	83 "
L.P. ahead -	114 "	96 "
L.P. astern -	76 "	72 "
Cruising - -	90 "	92 "

In the ahead and cruising turbines the dummy diameter is usually less than the drum, and the steam pressure acting on the small annulus so provided counterbalances (at certain speeds) the propeller thrust forward.

Trailing Shaft.—Collars are fitted in the length of tunnel shafting next the propeller shaft, on each side of the bearing, to act as a check on the longitudinal movement of the shafting, in the event of, say, a broken propeller or broken tail-end shaft. Should the propeller shaft break the loss of thrust in a forward direction might result in damage to the turbine by the steam thrust on the turbine blades aft; the check collar described will then bear up against the end of the tunnel bearing, and thus limit the end play of the shafting. A suitable clearance (from ½ in. to ¾ in. when cold), to allow of expansion when heated up under running conditions is allowed between the collar and faced-up end of the block, which is either of brass or of white metal.

Removable Thrust Rings.—When no adjustable coupling is fitted between the main and cruising turbines, the cruising thrust is fitted with loose horse-shoe type brass rings, which require to be taken out when running with the H.P. and L.P. turbines only. This is necessary owing to the danger of the unequal expansion between the main and cruising turbines "binding" the shaft when running ahead with the main turbines heated up and the cruising turbines cold.

Horse-Power of Turbines.

Shaft or Brake Horse-Power.—It is now well known that so far no method has been devised, or, in fact, is likely to be devised, for the indicating of the horse-power as done in the case of reciprocating engines, but the actual power transmitted along the shafting to the propeller may be determined by means of the "torsion meter" (pages 255-262), an instrument which measures the twist or torque put on the shaft by a given power. For accuracy of results it is advisable to have the shafting calibrated (page 133) beforehand, as different builds of shafts and materials give slightly varying results.

It should be noted that the shaft horse-power or brake horse-

power, as measured by the torsion meter, is the useful horse-power, and that the I.H.P. by comparison is a matter of indifference, the effective horse-power being actually transmitted along the shafting to the propeller being of chief importance.

Turbine Horse-Power.—The turbine horse-power may be found by calculation if the steam consumption, heat drop per pound steam, and turbine over-all efficiency are known, thus—

$$\text{Turbine Horse-Power} = \frac{\text{Heat drop} \times \text{Pounds steam per min.} \times 778 \times \text{Efficiency}}{33000}.$$

EXAMPLE.—The calculated heat drop per pound of steam is 290 heat units, and the steam flow is 1,300 lbs. per minute, determine the turbine horse-power if the turbine over-all efficiency is rated as 62 per cent.

$$\text{Then, Horse-Power} = \frac{290 \times 1300 \times 778 \times .62}{33000} = 5510.$$

Equivalent I.H.P.—Repeated trials have proved that the ratio o shaft horse-power by torsion meter as compared to indicated horse power is usually in the ratio of 90 to 100, or .9 to 1.

$$\text{Therefore, Equivalent I.H.P.} = \text{Shaft Horse-Power} \div .9.$$

EXAMPLE.—The collective shaft horse-power by torsion meter i found to be 8100; calculate the equivalent I.H.P.

$$\text{Then, equivalent I.H.P.} = 8100 \div .9 = 9000.$$

Turbine Diagram.—Heat drop and Pressure drop.

The diagram shown illustrates the successive variations in steam pressure and volu which occur throughout the H.P. and L.P. turbines. The figures given are, however, o approximate, and merely serve to indicate to the student the work done by the stea together with the corresponding changes in pressure, volume, &c., in its passage through t turbine casings.

DATA FOR DIAGRAM.

Coal per hour - - - - - -	7 tons.
Evaporation - - - - - -	8.5 lbs.
H.P. turbine initial pressure - -	140 lbs. gauge (dryness 1).
L.P. turbines „ „ - - - -	20 „ „ („ .92).
Condensers - - - - - -	27 in. vacuum („ .78).
Efficiency (assumed) - - - -	60 per cent.

NOTE.—Adiabatic expansion assumed.

NOTE.—The H.P. terminal pressure is approximately the L initial pressure, and the L.P. terminal pressures the condenser pressur

H.P. Turbines.

$$\text{Steam flow per min.} = \frac{7 \times 2240 \times 8.5}{60} = 2223 \text{ lbs.}$$

Assuming leakage loss as 10 per cent.—

$$\text{Then, Actual steam flow through turbines} = \frac{2222 \times 90}{100} = 1998 \Big\} \text{(Say, 20c lbs. per mi}$$

Initial, $140 + 15 = 155$ lbs. absolute,
and, 155 lbs. $= 361.1$ temperature and 859.6 units latent heat (from Table, page 7).
$361.1 + 461 = 822.1$ absolute temperature
Terminal, $20 + 15 = 35$ lbs. absolute.
and, 35 lbs. $= 259.3$ temperature and 931.6 units latent heat.
$259.3 + 461 = 720.3$ absolute temperature.

Heat drop per pound $= 859.6 \times 1 - 931.6 \times .92 + 822.1 - 720.3 = 104.33$ B.T.U.

$$\text{Horse-power} = \frac{104.33 \times 2000 \times 778 \times .60}{33000} = 2951.5.$$

Mean heat drop per blade row per pound steam $= 104.33 \div 48 = 2.17$ B.T.U.
Total pressure drop $= 155 - 35 = 120$ lbs.
Mean pressure drop per blade row $= 120 \div 48 = 2.5$ lbs.

L.P. Turbines.

Initial, 35 lbs. $= 720.3$ absolute temperature and 931.6 latent heat.
Terminal, 1.5 lbs. (absolute), 115.9 temperature, and 1033.2 units latent heat.
$115.9 + 461 = 576.9$ absolute temperature.

$$\text{Heat drop per pound} = 931.6 \times .92 - 1033.2 \times .78 + 720.3 - 576.9 = 194.58 \text{ B.T.U.}$$

$$\text{Horse-power (each L.P.)} = \frac{194.58 \times 1000 \times 778 \times .60}{33000} = 2752.4.$$

NOTE.—Each L.P. turbine receives *half* the total steam flow, or say 1,000 lbs. per minute.

Mean heat drop per blade row $= 194.58 \div 48 = 4.05$ B.T.U.
Total pressure drop $= 35 - 1.5 = 35.5$ lbs.
Mean pressure drop per blade row $= 33.5 \div 48 = .7$ lbs. (nearly).
Total horse-power $= 2951.5 + 2752.4 + 2752.4 = 8456.3$ (say 8456).

$$\text{Steam per hour per horse-power} = \frac{2000 \times 60}{8456} = 14.19 \text{ lbs.}$$

Coal per hour per horse-power $= 14.19 \div 8.5 = 1.6$.

NOTE.—Referring to Steam Table (page 3) the specific volume at 35 lbs. pressure absolute is 11.64 cub. ft., and at 1.5 lbs. absolute, 225.5 cub. ft., and these multiplied by the respective dryness fractions obtain the actual steam volumes, thus:—

$11.64 \times .92 = 10.7$ cubic feet.
$225.5 \times .78 = 176$,, ,,

Blade Tip Leakage Area.—The leakage over blade tips depends on the clearance between the rotor blades and the inside of the casing, and the clearance between the casing blades and the outside of the rotor. The so-called "blade height" includes the two working clearances referred to, and when these are deducted the effective heights of the blades are correspondingly reduced. To take an example:—

Rotor diameter outside, 48 in.
Blade height at 1st expansion, $1\frac{1}{16}$ in.
Tip clearance, $\frac{25}{1000}$ in., or .025 in.

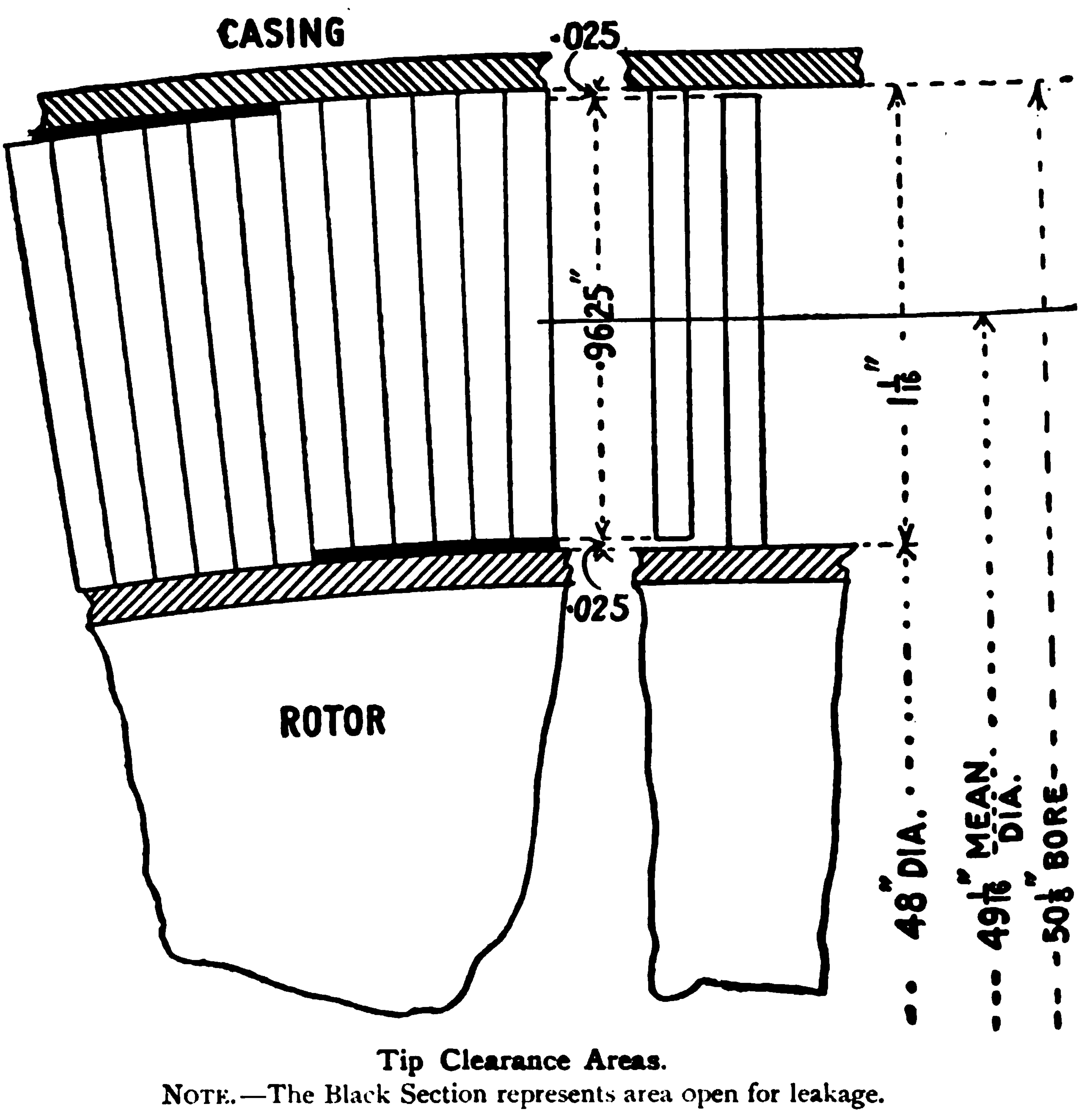
From the foregoing it will be obvious that the effective blade height is equal to $1\frac{1}{16}$ in. $-$ .025 in. $\times$ 2 in. $=$ 1.0125 in., and the *mean* diameter of blades 48 in. $+$ $1\frac{1}{16}$ in. $=$ $49\frac{1}{16}$ in. To find the per cent. leakage over blade tips, if the blade openings at exit edge are equal to .3 of annular area formed by blades, then—

$$49.0625 \times 3.1416 \times .9825 \times .3 = 45.4 \text{ sq. in. clear area,}$$

$$\text{and } 49.0625 \times 3.1416 \times .025 \times 2 = 7.706 \text{ sq. in. leakage area.}$$

$$\text{Therefore, } \frac{7.706 \times 100}{45.4} = 16.9 \text{ per cent.}$$

NOTE.—$49\frac{1}{16}$ in. $=$ 49.0625 in.



Tip Clearance Areas.

NOTE.—The Black Section represents area open for leakage.

The tip clearance area is therefore equal to about 17 per cent. of the steam area through blades.

Number of Rows of Blades and Blade Heights.—The number of rows of blades, according to the formula given by Mr E. M. Speakman on page 23, works out as follows:—

EXAMPLE 1.—Calculate the required number of rows of blades of a turbine if the H.P. blade velocity is to be 100 feet per second.

RULE.—$1500000 = V_t^2 \times$ number of rows.

Therefore, $\dfrac{1500000}{V_t^2} =$ number of rows, and $\dfrac{1500000}{100^2} = 150$ rows of blades.

NOTE.—By this is meant that if the H.P. turbine develops the total power, 150 rows of blades will be required in the rotor and on the casing of that turbine, but as the H.P. develops only about one-third of the total power (with the standard 3 ahead turbine arrangement) then, $150 \div 3 = 50$ rows of blades for the H.P. turbine actually required. The same statement holds good for the L.P. turbines, only as the two L.P. turbines develop two-thirds of the total power, the result must be multiplied by this fraction.

EXAMPLE 2.—Calculate the required number of blade rows required in each L.P. turbine referring to Example No. 1, the L.P. blade velocity being 140 feet per second.

Then, $\dfrac{1500000}{140^2} = 76$, and $76 \times \frac{2}{3} = 50$ blade rows for each L.P. turbine.

EXAMPLE 3.—The total number of rows of blades in the turbines of a channel steamer is 192. Calculate the mean velocity (V_t) of the blades.

Then, $1500000 = V_t^2 \times$ number of rows.

Therefore, $1500000 = V_t^2 \times 192$,

and $\dfrac{1500000}{192} = V_t^2 = \dfrac{1500000}{192} = 7812$;

so that, $\sqrt{7812} = 88$ feet per second $= V_t$.

By the rule given, determine the number of blade rows required in the H.P. and L.P. turbines of the "Lusitania" and "Mauretania," given that the rotor drum diameters are 96 in. and 140 in., the revolutions 180 per minute, and the constant adopted 1400000.

Then, surface velocity of H.P. drum per sec. $= \dfrac{96 \times 3.1416 \times 180}{12 \text{ in.} \times 60 \text{ min.}} = 75.4$ ft.,

and, number of blade rows required for full work of steam $\Bigr\} = \dfrac{1400000}{75.4^2} = 246$ rows of blades.

As, however, the H.P. is designed for only half the work of the steam passing through it, the L.P. to develop the other half, $\Bigr\}$ then, $246 \div 2 = 123$ rows of blades actually required.

Again, surface velocity of L.P. drum per sec. $= \dfrac{140 \times 3.1416 \times 180}{12 \text{ in.} \times 60 \text{ min.}} = 110$ ft.,

and, $\dfrac{1400000}{110^2} = 115$ blade rows for full work.

Therefore, $115 \div 2 = 57$ blade rows for half work.

NOTE.—The actual blade rows fitted are: H.P. 123, and L.P. 56 which agree closely with the foregoing.

Initial Steam Velocity.—According to an article in *Engineering* of 13th December 1907, on "The Practical Proportioning of the Reaction Steam Turbine," the velocity of the H.P. turbine initial steam is determined by the following:—

$$\text{Rule, Velocity of steam at 1st H.P. expansion} = \frac{\text{Constant}}{\sqrt{N}}$$

Where Constant = 2700 (64 steam expansions assumed).
" " = 3000 (more than 64 expansions assumed).
" N = No. of blade rows in turbines from rule given on page 23.

The successive blade height ratios is given as $\sqrt{2}$, and the approximate ratio of H.P. to L.P. rotor drum also as $\sqrt{2}$, but these ratios vary somewhat in different cases, as may be tested by working out and comparing with the examples from actual practice, pages 138 to 218. To apply the foregoing to turbines as actually constructed, take the following cases:—

EXAMPLE 1.—No. of blade rows 166, and constant 2,700 for 64 expansions.

$$\text{Then, } \frac{2700}{\sqrt{166}} = \frac{270.0}{12.88} = 210 \text{ ft. (nearly) per second.}$$

From this the required clear area through blades, and therefore blade heights, may be calculated, given the steam flow per second, the actual volume of the steam, and the rotor drum diameter.

EXAMPLE 2.—Calculate the initial velocity of the steam at the H.P. turbine of the U.S.S. "Chester," given that the blade rows = 140 and the constant adopted = 2,700.

$$\text{Then, velocity} = 2700 \div \sqrt{140} = 2700 \div 11.83 = 228 \text{ ft. per second.}$$

EXAMPLE 1.—Determine the required blade height of the 1st L.P. expansion of the "Dreadnought" turbines, given that the rotor drum diameters are 68 in. H.P., and 92 in. L.P., the last H.P. expansion blades being 5 in. and the blade height increase, to allow of steam expansion, to be in the usual ratio of $\sqrt{2}$ times.

Then, Ratio of rotor diameters $= 92 \div 68 = 1.35$.

For similar rotor diameters the 1st L.P. expansion blade height would now require to be equal to $5 \times \sqrt{2} = 7.05$ in. But as the L.P. rotor diameter is 1.35 times that of the H.P. rotor, then, $7.05 \div 1.35 = 5.22$ in. height of blade, neglecting increase of steam volume and velocity, which is also proportional to the increase of rotor diameters; therefore $5.22 \div 1.35 = 3.8$ in. or $3\frac{7}{8}$ in. blade height, which is the actual size adopted.

EXAMPLE 2.—Calculate the required blade height of the 1st L.P. expansion, given that the rotor drum diameters are 30 in. H.P. and 45 in. L.P., and the last H.P. expansion blade height $4\frac{1}{2}$ in. The turbines are three in number, one H.P. and two L.P.

Then, $4.5 \times \sqrt{2} = 6.36$ in. blade height for similar diameter of rotor.
and, ratio of rotor diameters $= 45 \div 30 = 1.5$.

Therefore, $6.36 \div (1.5 \times 1.5) = 2.82$ in. blade height if one L.P. turbine only is fitted, so that, $2.82 \div 2 = 1.41$ (say 1.5 in.) blade height for each L.P. turbine.

EXAMPLE 3.—Calculate the blade heights at each expansion, given the following:—Horse-power, 10,000; initial pressure, 140 lbs. by gauge (volume, 2.87 cub. ft.); steam consumption (estimated), 14 lbs. per horse-power hour; H.P. rotor drum, 48 in. diameter; L.P. rotor drum, 68 in. diameter; revolutions (designed), 500 per minute; blade rows 144, and constant adopted for initial steam velocity, 2,700. Assume normal blades giving a clear area ratio of one-third blade annulus.

H.P. Turbine.

Initial velocity of steam $= 2700 \div \sqrt{144} = 225$ ft. per second.

$$\text{Steam flow per second} = \frac{10000 \times 14 \times 2.87}{60 \times 60} = 111.61 \text{ cub. ft.}$$

Required clear area $= 111.61 \div 225 = .496$ sq. ft.

Required annulus area $= .496 \times 3 = 1.488$ sq. ft.,

and, $1.488 \times 144 = 214.272$ sq. in.

$$\text{Therefore, diameter across blade tips} = \sqrt{\frac{214.272 + 48^2 \times .7854}{.7854}} = 50{\cdot}76,$$

$$\text{and, } \frac{50.76 - 48}{2} = 1.38 \text{ blade height at 1st expansion.}$$

EXAMPLE.—To find the blade height at the 2nd, 3rd, and 4th expansions—

Then, $1.38 \times 1.42 = 1.95$ in., say 2 in. at 2nd expansion.
$2 \times 1.42 = 2.84$ in., „ $2\frac{3}{4}$ in. „ 3rd „
$2.75 \times 1.42 = 3.9$ in., „ 4 in. „ 4th „

NOTE.—Assuming that $\sqrt{2} = 1.42$.

L.P. Turbine.

Drum ratio $= 68 \div 48 = 1.4166$.

$$\text{Therefore blade height of 1st expansion} = \frac{4 \times 1.42}{1.4166 \times 1.4166 \times 2} = 1.4, \text{ say } 1\tfrac{3}{8} \text{ in.}$$

Blade heights at the other expansions—

1.375 in. $\times 1.42 = 1.95$ in., say 2 in. at 2nd expansion.
2 in. $\times 1.42 = 2.84$ in., „ $2\frac{3}{4}$ in. „ 3rd „
2.75 in. $\times 1.42 = 3.9$ in., „ 4 in. „ 4th „
4 in. $\times 1.42 = 5.68$ in., „ $5\frac{1}{2}$ in. „ 5th „
5.5 in. $\times 1.42 = 7.8$ in., „ 8 in. „ 6th „
"semi-wing" blades 8 in. „ 7th „
„ „ „ 8 in. „ 8th „

NOTE.—The blade heights as shown are measured from *above* the surface of the drum only, the actual blade height over all being in excess of these by the depth of the fitting groove, which varies from $\frac{1}{4}$ in. in small H.P. blades to $\frac{3}{8}$ in. or $\frac{1}{2}$ in. in large L.P. blades. It should also be noted that the tip clearance is neglected in the above dimensions, which if deducted would reduce the blade heights by about $\frac{30}{1000}$ in. in the H.P. blades and about $\frac{50}{1000}$ in. in the L.P. blades.

In the "Lusitania" and "Mauretania" the increase of blade height is in the ratio of 1.22 (full) in the H.P. turbines and 1.27 (full) in the L.P. turbines; whereas in the "Invincible" class the ratios are 1.33 for H.P. blades and about 1.4 (full) for L.P. blades. The rotor drum ratio of the Cunarders is as 1 : 1.45 (full) and of the "Invincible" class as 1 : 1.25.

EXAMPLE 1.—Given that the blade height at the 1st H.P. expansion of the "Lusitania" is $2\frac{3}{4}$ in., calculate the blade heights of the 2nd, 3rd, 4th, 5th, 6th, 7th, and 8th H.P. expansions.

Then,	$2.75 \times 1.22 = 2.355$,	say $3\frac{3}{8}$ in.	at 2nd expansion.
	$2.375 \times 1.22 = 4.117$,	" $4\frac{1}{4}$ in.	" 3rd "
	$4.25 \times 1.22 = 5.185$,	" $5\frac{1}{4}$ in.	" 4th "
	$5.25 \times 1.22 = 6.405$,	" $6\frac{1}{2}$ in.	" 5th "
	$6.5 \times 1.22 = 7.930$,	" 8 in.	" 6th "
	$8 \times 1.22 = 9.76$,	" 10 in.	" 7th "
	$10 \times 1.22 = 12.20$,	" $12\frac{1}{4}$ in.	" 8th "

EXAMPLE 2.—Given that the 1st expansion blade height is $8\frac{1}{4}$ in., calculate in the same manner the blade heights at the 2nd, 3rd, 4th, and 5th L.P. turbine expansions, the remaining expansions (6th, 7th, and 8th) being of the special "winged" type.

Then,	$8.25 \times 1.27 = 10.477$,	say $10\frac{1}{2}$ in.	at 2nd expansion.
	$10.5 \times 1.27 = 13.335$,	" $13\frac{1}{2}$ in.	" 3rd "
	$13.5 \times 1.27 = 17.145$,	" $17\frac{1}{4}$ in.	" 4th "
	$17.25 \times 1.27 = 21.907$,	" 22 in.	" 5th "

NOTE.—This height of 22 in. also holds good for the 6th, 7th, and 8th expansions, but the blade openings are increased much in excess of the normal amount of .33 of the annulus area by these blades being of "semi-wing," "wing," and "double-wing" section and angles, which, in the case of the last named, almost brings the blades parallel to the drum in a fore-and-aft line.

EXAMPLE.—Calculate the blade heights of the 1st H.P. expansion of the "Indomitable," given that the drum is 92 in. diameter, the revolutions 250 per minute, the horse-power 40,000, and assuming 14.2 lbs. of steam per horse-power hour. Assume an initial pressure of 140 lbs. (gauge), and take number of rows as 120, with a constant of 2,700.

Then, Velocity of initial steam flow per sec. $= \frac{2700}{\sqrt{120}} = \frac{2700}{11} = 245$ ft.,

and, Steam flow per sec. $= \frac{20000 \times 14.2 \times 2.87}{60 \times 60} = 226.4$ cub. ft.

NOTE.—20,000 horse-power for each pair of H.P. and L.P. turbines.

„ 140 lbs. gauge = 155 lbs. absolute = 2.87 cub. ft. specific volume. See "Steam Table," page 3.

Clear area required $= 226.4 \div 245 = .924$ sq. ft.

Annulus area required $= .924 \times 144 \times 3 = 399$ sq. in.

Therefore, Diam. across blade tips $= \sqrt{\frac{399 + (92^2 \times .7854)}{.7854}} = 94.72$ in.,

and, Blade heights $= \frac{94.72 - 92}{2} = 1.36$ in., say $1\frac{3}{8}$ in.

EXAMPLE.—Also proceed to determine the blade heights of the 2nd, 3rd, 4th, 5th, and 6th H.P. expansions of the same turbine, the increase of height being in the ratio of 1.33 (full).

Then, $1.375 \times 1.33 = 1.828$, say $1\frac{7}{8}$ in. at 2nd expansion.
$1.875 \times 1.33 = 2.493$, „ $2\frac{1}{2}$ in. „ 3rd „
$2.5 \times 1.33 = 3.325$, „ $3\frac{1}{2}$ in. „ 4th „
$3.5 \times 1.33 = 4.655$, „ $4\frac{3}{4}$ in. „ 5th „
$4.75 \times 1.33 = 6.317$, „ $6\frac{3}{8}$ in. „ 6th „

NOTE.—In the foregoing the number of blades in each expansion will probably vary from about 16,000 in the 1st to about 9,000 in the 6th of the rotor, and from about 11,000 in the 1st to about 7,000 in the 6th of the casing.

EXAMPLE.—Calculate (1) the area open for steam flow; (2) the steam flow in cubic feet per second; (3) in lbs. per second; and (4) the horse-power developed, allowing 15 lbs. steam per horse-power hour, for the following ("Lusitania"):—The H.P. rotor drum being 96 in. diameter; blade height at 1st expansion, $2\frac{3}{4}$ in.; blade opening at exit edges, .09 in.; revolutions, 180 per minute; initial pressure, 180 lbs. gauge; and assuming a mean blade speed to steam speed ratio of .42 at this expansion. Assume the number of blades per row to be 1000, and neglect blade tip clearance.

Then, Mean diameter across blades $= 96$ in. $+ 2.75$ in. $= 98.75$.

Mean blade speed in ft. per second $= \frac{98.75 \times 3.1416 \times 180}{12 \times 60} = 77.5$ ft.

Steam speed per second $= 77.5 \div .42 = 184$ ft.

Area of exit opening between each pair of blades $= .09 \times 2.75 = .2475$ sq. in.

„ „ „ per ring of blades $= \frac{.2475 \times 1000}{144} = 1.718$ sq. ft.

Cub. ft. steam flow per second $= 1.718 \times 184 = 316.11$ cub. ft.

Pounds steam flow „ „ $= 316.11 \times .432 = 136.55$ lbs.

„ „ „ hour $= 136.55 \times 60 \times 60 = 491580$ lbs.

Horse-power (one side) $= 491580 \div 15 = 32772$.

Horse-power (both sides) $= 32772 \times 2 = 65544$.

NOTE.—185 lbs. + 15 = 195 lbs. absolute, and 195 absolute = .432 density, from "Steam Table," page 3.

NOTE.—The actual area of opening as given between blades for steam flow is less than the third usually assumed for normal blades, as the following calculations prove:—

$$\text{Blade annulus (clearance neglected)} = \frac{(96+2.75)\times 3.1416 \times 2.75}{144} = 5.92 \text{ sq. ft.}$$

$$\text{Ratios of actual opening to annulus} = 1.718 \div 5.92 = .2902.$$

The openings between blades can be varied during construction by means of "closing up" and "opening out" tools for the purpose which change the blade exit angle as considered advisable. This work is occasionally done after trials, and in some cases has produced decided improvement in the results (see page 44). It might be argued from this that a certain amount of "trial and error" method is still required in turbine work to obtain the best results.

Diameters of Rotors and Dummies.—The following figures give the respective sizes of the rotor drums and dummies as usually designed for the turbines of channel steamers:—

Turbine.	Diameter of Rotor Drum.	Diameter of Dummy.
H.P. - -	5 feet 3 inches.	5 feet 0 inches.
L.P - -	7 „ 6 „	6 „ 8 „
Reverse -	5 „ 10 „	5 „ 9 „

NOTE.—The steam pressure acting on the annulus area formed by the difference of drum and dummy diameter together with the effective combined steam pressure on the blades is designed to balance the propeller speed at maximum speed, although at lower speeds the propeller thrust may either more or less than balance this pressure.

Water Condensed in Turbines.—Starting with dry saturated steam at the initial end of the H.P. turbine, a mixture of condensed steam and water is produced at the last L.P. expansion due to the (partial) adiabatic expansion of the steam in doing work. This must not be confused with the initial condensation which occurs in reciprocating engine cylinders, and which is due to the alternate heating up and cooling down during admission and exhaust.

A small amount of condensation takes place in the H.P. turbine due to the decrease of dryness fraction, but much more occurs in the L.P. turbines, which accounts for the "wet" air pump connection to the after end of these turbines.

According to the "error" diagram of Mr E. M. Speakman (page 21), the actual volume of the steam at a pressure of 2 lbs. absolute is

143.4 cub. ft. per lb., which gives a dryness fraction of .833. If then the steam flow be known, the condensed water produced can be found as follows :—

Water condensed in turbines per minute = steam flow per minute × (1 - .833) (assuming a terminal L.P. pressure of 2 lbs. absolute).

EXAMPLE.—H.P. initial pressure, 150 lbs. gauge; H.P. terminal pressure, 22 lbs. gauge; and L.P. initial pressure, 20 lbs. gauge.

Horse-power, 16,000, and assume 14 lbs. steam per horse-power hour. Assume H.P. initial steam of dryness 1.

According to the "error" diagram of Mr Speakman, 35 lbs. pressure absolute = .932 dryness factor and 2 lbs. pressure = .835 dryness factor (page 41).

NOTE.—20 + 15 = 35 lbs. absolute.

Then, total weight of water condensed in turbines $\Big\} = 16000 \times 14 \times (1 - .835) = 36960$ lbs. per hour.

Again, weight of water condensed in H.P. turbine $\Big\} = 16000 \times 14 \times (1 - .932) = 15232$ „ „

Weight of water condensed in each L.P turbine $\Big\} = \dfrac{16000 \times 14}{2} \times (.932 - .835) = 10864$ lbs. „

As mentioned elsewhere, most of the water condensed in the H.P. turbine is blown out with the steam into the two L.P. turbines, from which it is drawn off by the wet air pumps. These pumps are kept working when the turbines are stopped temporarily, in addition to working continuously when running.

Power Developed by each Turbine.—(*A.*) Referring to Example No. 16, page 173, it will be seen that, running at a trial speed of 14.73 knots, the pressures indicated were as follows :—

H.P. turbine - - - - -	35 lbs. (gauge).
L.P. turbines - - - - -	12 in. vacuum.
Condenser - - - - - -	27 in. „

Assuming that the power varies as the cube of the speed, and as 9,000 horse-power gives 22 knots,

$$\text{Then, } \frac{14.73^3 \times 9000 \text{ I.H.P.}}{22^3} = 2701 \text{ Horse-Power.}$$

As there are three shafts, 2701 ÷ 3 = 900 Horse-Power developed by each turbine; or, 900 × 33,000 = 29,700,000 foot-pounds of effective work per minute.

This energy is given out by each L.P. turbine in working between an initial pressure of 9 lbs. absolute and final pressure of about 2 lbs. absolute, or 9 − 2 = 7 lbs. pressure drop in all, and this total pressure drop of 7 lbs. gives a heat drop sufficient to obtain 900 horse-power.

As, 29700000 foot-pounds ÷ 778 = 38174 Units Heat Drop per minute.

If each L.P. turbine is made up of, say, 64 rows of blades in all, then the mean pressure drop per row is equal to .10 of a pound, as

$7 \div 64 = .10$; but the actual pressure drop is of a gradually decreasing quantity from the initial to the exhaust end of each turbine.

(*B.*) Again, taking the lowest trial speed of 10.68 knots, we find that the power required amounts to

$$\frac{10.68^3 \times 9000}{22^3} = 1030 \text{ Horse-Power.}$$

And as this is developed by the three shafts combined,

Then, $1030 \div 3 = 344$ Horse-Power per shaft or turbine,
and, $344 \times 33000 = 1032000$ foot-pounds of work per minute.
Therefore, $1032000 \div 778 = 1326$ units effective Heat Drop per minute.

The L.P. turbines are working in a vacuum of 21 in. at the initial end and 28 in. at the exhaust end. This heat drop is therefore given out by each L.P. turbine in working between pressures of $4\frac{1}{2}$ lbs. initial and $1\frac{1}{2}$ lb. final pressure (absolute), or within a total pressure drop of $4\frac{1}{2} - 1\frac{1}{2} = 3$ lbs.

NOTE.—21 in. $\div 2 = 10.5$ lbs., and 15 lbs. atmospheric pressure $- 10.5 = 4.5$ lbs. absolute.

Therefore, $3 \div 64$ Rows $= .046$ of a pound pressure drop per Row.

The effective heat value of low-pressure steam as utilised in turbines will perhaps be apparent from the foregoing figures, which are taken from actual practice.

To sum up—

In case (*A*) the H.P. turbine develops 900 horse-power working between pressures of 50 lbs. absolute and 9 lbs. absolute. So that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $50 - 9 = 41$ lbs.

Each L.P. turbine also develops 900 horse-power, working *in a vacuum*, or between pressures of 9 lbs. absolute and 2 lbs. absolute, so that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $9 - 2 = 7$ lbs.

In case (*B*) the H.P. turbine develops 344 horse-power working between pressures of 25 lbs. absolute and $4\frac{1}{2}$ lbs. absolute, so that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $25 - 4.5 = 20.5$ lbs.

Each L.P. turbine also develops about 344 horse-power, working *in a vacuum*, or between pressures of $4\frac{1}{2}$ lbs. absolute and $1\frac{1}{2}$ lb. absolute, so that the total pressure drop is 3 lbs., as $4.5 - 1.5 = 3$ lbs. The necessary heat drop is therefore obtained by this very small pressure drop, and the kinetic energy thus given up develops the required power.

SECTION II.

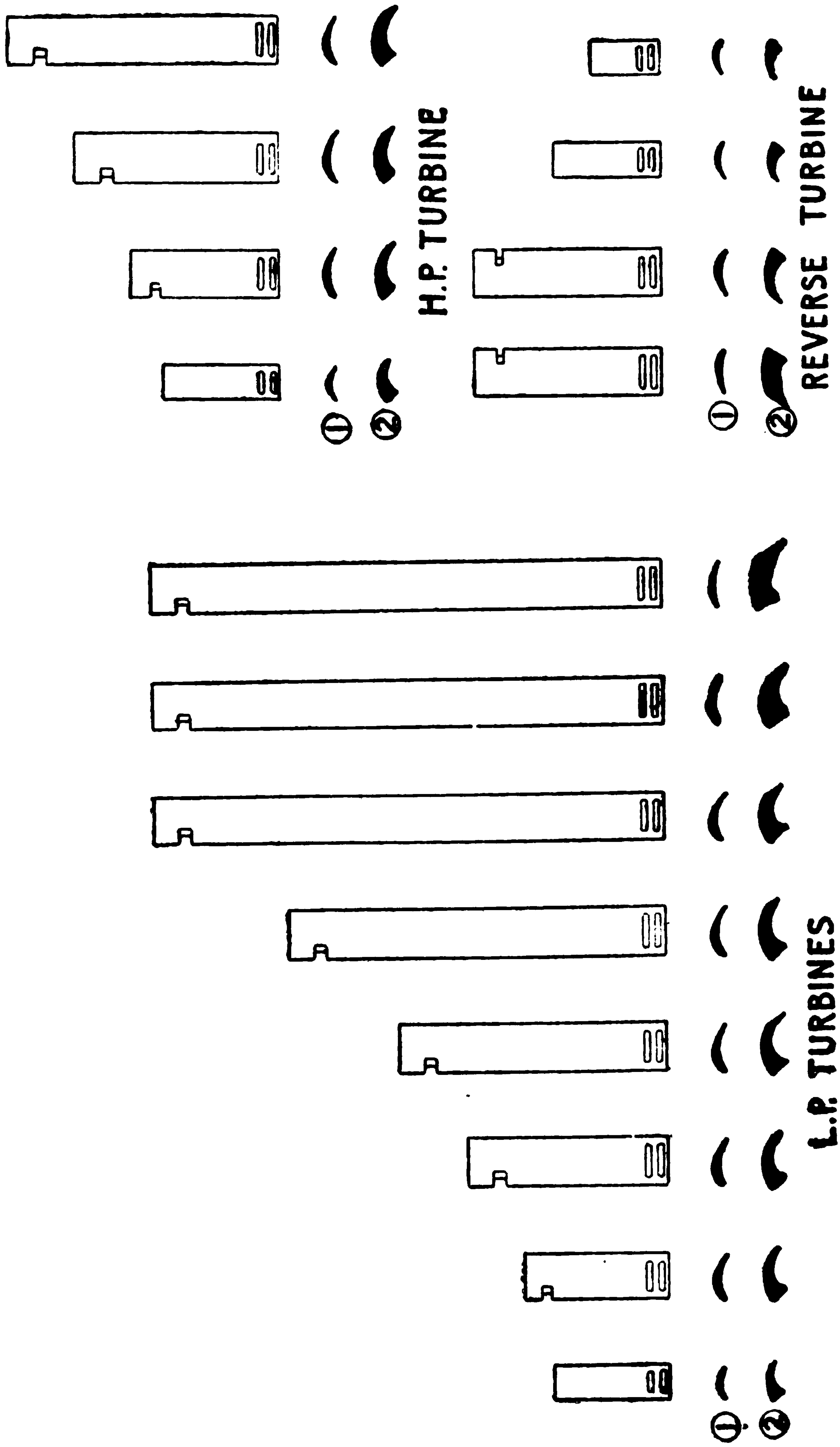
WORKSHOP PRACTICE.

THE turbine proper consists of two main parts; these are called the casing or cylinder, and the rotor.

The rotor is the revolving part, the casing or cylinder being stationary: the rotor consists of a round steel drum, into each end of which are inserted steel wheels or centres. Through the centre of these wheels spindles are fitted, which form the bearing shaft for the rotor. The outside of the drum is turned by machine, and at given distances grooves are cut, into which the brass blades are inserted. The casing is a cast-iron cylinder, which is turned inside, and is of increasing diameter at given distances in its length; grooves are also cut in the casing, and into these grooves brass blades of similar construction are inserted. The steam is admitted at the small end of the casing and passes through the blades. The reason for the increasing diameter of the casing is that the steam, after having done work, drops in temperature and pressure and increases in volume, so that to have a perfect arrangement of blades every row should be longer than the preceding one. In practice this has not been found practicable, and the blades are arranged in groups of different lengths and spacing, each group being termed an "expansion."

The blades do not lie parallel to the shaft axis or at right angles to it, but are placed at a slight angle, the shaft blades being at the opposite angle to those of the casing (see sketch). Each turbine *begins* with a ring of casing blades, and *ends* with a ring of rotor blades.

Usually in a high-pressure turbine there are 4 expansions, with 12 to 16 rows in each expansion, and in a low-pressure turbine there are usually 7 or 8 expansions of 6 to 10 rows in each. The blades in the rotor project outwards between the rows of blades in the casing, and the blades of the casing project inwards between the blades of the rotor. As explained elsewhere, the steam is admitted at the small end of the casing, and first passing through a row of casing blades is deflected so as to strike the rotor blades: after passing through the first row of moving blades it is again deflected, and strikes on the next row of fixed blades, which again deflects the steam so that it



Complete Set of Blades and Packing Pieces (half size).

1. Plan or edge view of Blades. 2. Plan of Packing Pieces.

NOTE.—The above are from the turbines of a steamer of about 5,000 I.H.P.

impinges on the next row of rotor blades, the steam reacting from one row to the other and flowing from the steam admission end of the turbine to the exhaust end. There are usually one high-pressure and two low-pressure turbines fitted. These turbines drive three shafts. The steam is admitted to the high-pressure turbine and exhausts from it into the two low-pressure turbines. Attached to the low-pressure turbines are the reverse turbines, which are used for going astern.

The economical running of turbines is greatly dependent upon the reduction of the clearance spaces to the utmost limit compatible with working conditions; this, of course, applies more or less to all steam engines, but in the case of turbines the steam blows continuously through from end to end over the tips of the blades, and this necessarily results in considerable loss of steam which, needless to say, is unavoidable.

TABLE III.—STANDARD BLADING DIMENSIONS.

Height (H)	1"	2"	3"	4"	6"	8"	10"	12"	15"	18"	21"	24"	30"
Width (W)	3/8"	3/8"	1/2"	1/2"	5/8"	5/8"	5/8"	3/4"	7/8"	1"	1"	1 1/4"	1 1/2"
Pitch (P)	1 1/8"	1 1/8"	1 1/4"	1 5/8"	1 3/4"	2 1/4"	2 1/4"	2 1/2"	2 5/8"	3 1/8"	3 1/4"	3 3/8"	4"
Axial clearance (C) ...	3/16"	3/16"	1/4"	5/16"	3/8"	7/16"	1/2"	1/2"	9/16"	9/16"	5/8"	11/16"	7/8"

NOTE.—While the above represents general practice, it is obvious that such a table is largely arbitrary.

Blade Dimensions

Construction of Rotors

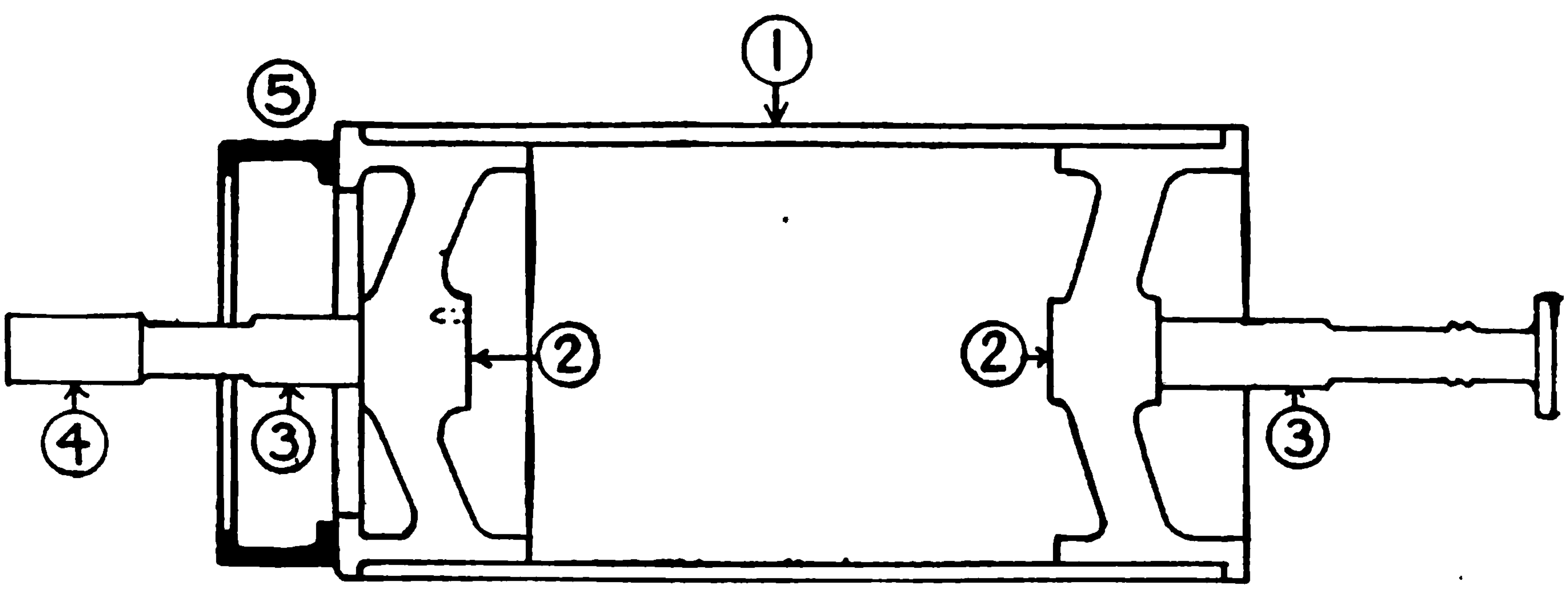
The rotor consists of—

1st, the drum.
2nd, the wheels.
3rd, the shafts or spindles.
4th, the dummy.

In large rotors, the drums being of great length are usually in two pieces, and have a centre wheel at the joint of the drum. The description which follows is of an H.P. rotor of the usual design of construction up to about 18,000 horse-power. The L.P. rotor will be described later.

The drums are worked out of the solid ingot, and are rolled out in large rolls. It may be mentioned that for the manufacture of

drums of large diameter there are only a few firms in the country who can undertake this work. The drums being delivered at the works rough turned, the next operation is the boring out of the inside to receive the wheels ; this is usually done either in a lathe when the drum is small, or in a large boring machine when the drum is of large dimensions. When binding the drum either in the lathe or machine care must be taken to have an equal pressure at the different points where the drum is held so as to ensure that no distortion takes place. To overcome this difficulty it is advisable to have two cast-iron half rings or glands lined with wood ; the wood is bored out to the size of the outside of drum, and then the drum is put inside bottom half of ring, top half put on, and screwed up.

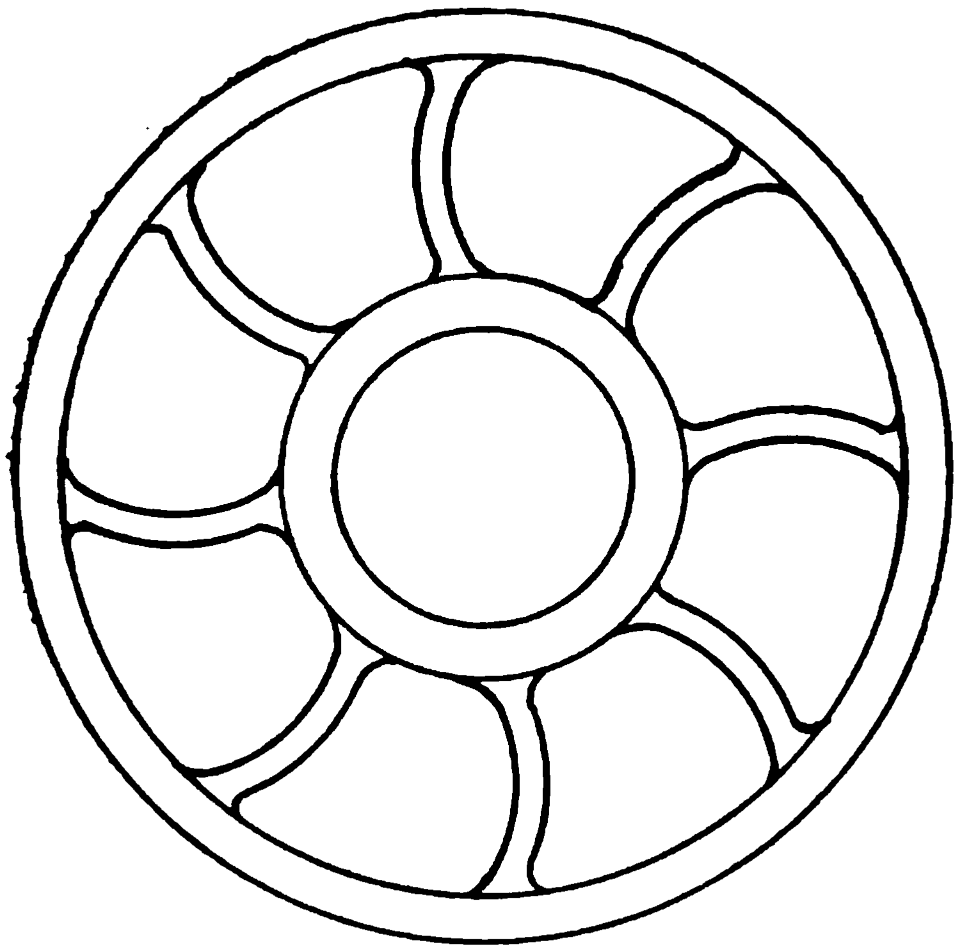


H.P. Rotor Drum.
1. Drum. 2. Wheels. 3. Spindles.
4. Thrust. 5. Dummy.

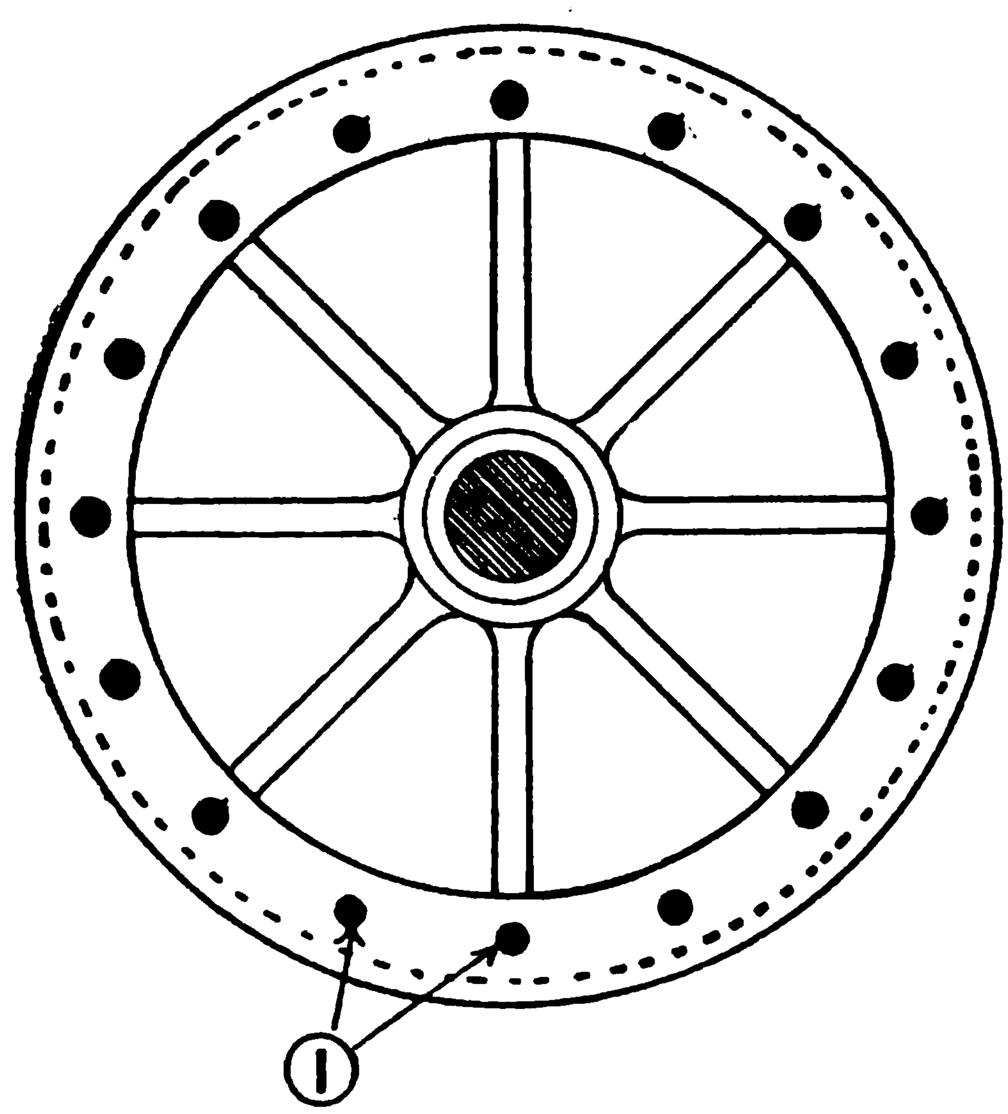
The drum being bored, the wheels are now brought forward. The wheels are of steel, either cast or forged, and there are several different designs, usually the cast-steel wheels have a number of ribs, and these occasionally are of unequal number so as not to throw a transverse strain on the wheel when cooling out after casting. Another type of wheel is of the forged steel type, of disc form, forged from the ingot and turned all over, this ensuring a perfect balance. It may be mentioned that this type of wheel was adopted in the " Mauretania." In large wheels of the cast type the outer rim is sometimes cast with three spaces, or joints in the outer rim, so as to avoid contraction cracks. These joints are afterwards filled up with steel liners bolted to a flange on the wheels.

The wheels are bored out in the centre or nave to receive the shafts or spindles, which are turned on the ends, to sizes suitable for shrinking.

After wheels and shafts are machined the wheels are put into a drying stove in the foundry, if there is no suitable stove in the erection shop for this work, as it is advisable that the wheels should be equally heated up all over. The wheel is then heated up and expanded



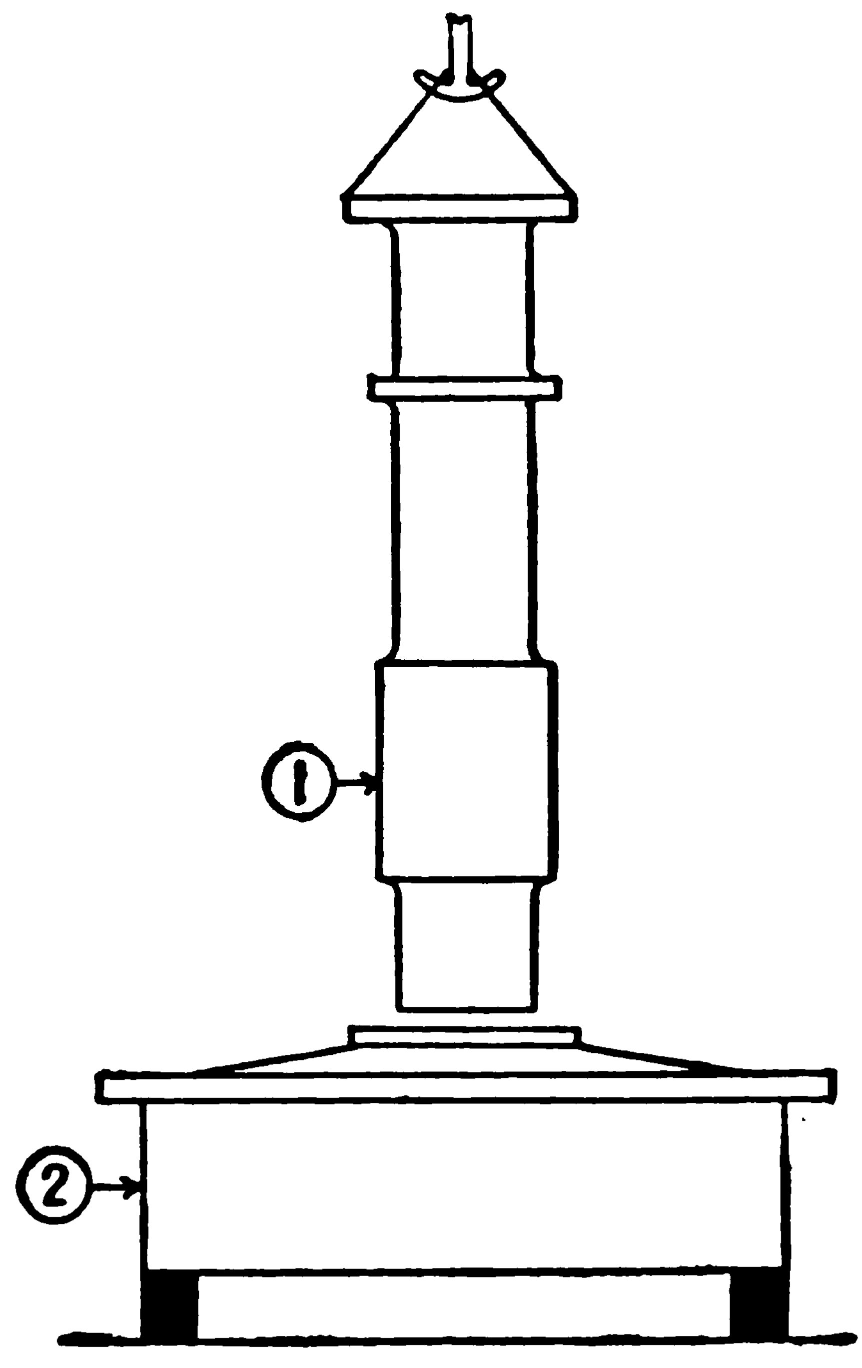
"Wheel" or "Centre."
With Curved Arms to allow of Expansion when Heated.



"Wheel" or "Centre."
1. Bolt Holes for Dummy.

to $\frac{1}{32}$ in. larger than the diameter of the shaft which is to go into it. The spindle must be slung dead plumb, and care should be taken to have the wheel also truly levelled up.

After taking wheel out of stove or furnace the spindle is suspended above and gently lowered until it enters wheel, then lowered as smartly as possible until spindle rests on shoulder. In cooling out the wheel it is preferable to use jets of compressed air in preference to water, and to cool down part of wheel nearest shoulder on spindle first so as to ensure the shoulder being close to wheel, as in contracting while cooling this may sometimes be open. Compressed air for cooling is preferable to water for the reason that it does not run down the wheel, and only cools out the part intended. The wheel being



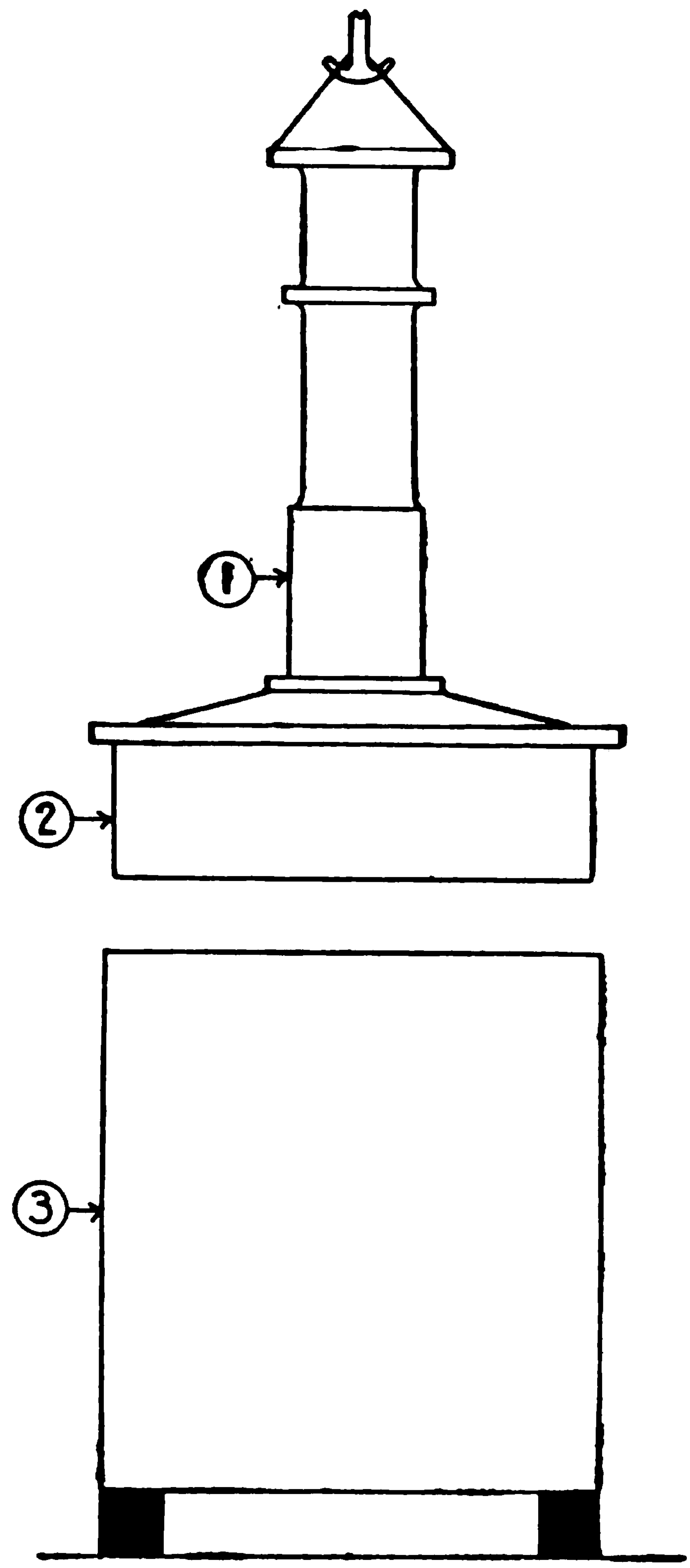
Method of Shrinking on Spindles to Wheels.

1. Spindle. 2. Wheel.

cooled is now put into machine and holes bored half in shaft and half in wheel. Pins are screwed into these holes, the heads of pins being riveted over to prevent slackening back.

In large rotors, instead of screwed pins, round driving keys are fitted, and the end of the spindle is screwed and a round nut put on which covers keys. The nut is prevented from slackening back by stop pins being fitted half into shaft and half into nut. The wheels are now put into lathe again and outside diameters turned to suit inside bore of rotor drum, and the same amount as before allowed for shrinkage. The dummy is a steel ring which is attached to the steam inlet end of

the rotor. There are two kinds of dummies, facial and radial. Facial dummies are usually adopted on the low-pressure and high-pressure



Method of Shrinking on Wheel to Drum.

1. Spindle. 2. Wheel. 3. Drum.

NOTE.—The wheel is shown suspended in readiness for lowering into the drum which has been heated and expanded to receive it.

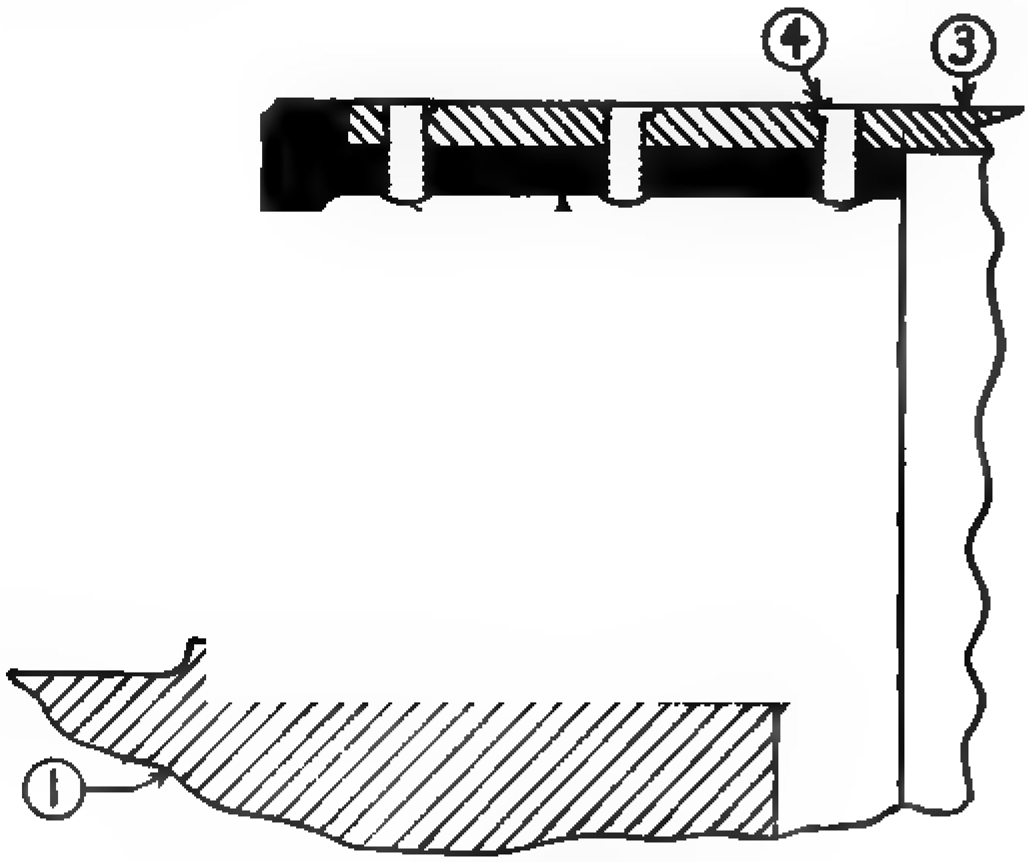
ahead turbines, the radial dummy is usually fitted on the astern turbine.

The use of the dummy is to prevent the steam leaking to the

glands at the end of the casing, and also to prevent the steam passing through the interior of the rotor to the exhaust, instead of doing legitimate work by passing through the turbine blades.

Shrinking or Building of Rotors.

The rotor drums are erected, if the rotors are small, on suitable blocks; if large, then it is necessary to have a shrinking pit so that the spindle of rotors will clear when turned up on end. The drums are set up and plumbed and then heated with either blow lamps in the case of small rotors, or by gas rings when constructing large rotors,



Method of Connecting Wheel, Spindle, and Drum.
1. Spindle. 2. Wheel. 3. Rotor Drum. 4. Screwed and Riveted Pins, with small Feather to Prevent Slackening Back.

the drum being heated and expanded so that a gauge $\frac{1}{32}$ in. larger than the wheel will pass clear down to a little further than the depth of the wheel. The wheel and spindle are slung, care being taken to ensure them being plumb, and the wheel is entered into drum and lowered into place.

The drum is now cooled out at the top first in the same manner as before by compressed air. The drum is now marked off for the rivets or pegs which bind it to the wheel; the rivets are pitched zig-zag, a few holes are bored, and temporary bolts put into same, after which the drum is turned upside down with the rotor shaft either in the pit or, if a small rotor is being constructed, between wood blocks. The other wheel is now shrunk in like manner, bolts or pins put in

and rotor canted down. The rotor now goes to boring machine for the cutting of the rivet boles, after which rivets are put in and closed up. The rotor is now ready for turning. After rotor is put in lathe care should be taken to see that it is true, and if not the centre should be adjusted to make it so. It is then turned all over the body to the required given diameter and marked off for grooving. This operation

Hollow Cast Turbine Wheel (Naval type).

1. Steam Inlet. 2. Dummy. 3. Rotor Drum. 4. Hole to allow Admission of Steam to Hollow Spindle for Uniform Heating up and Expansion. 5. Hollow Spindle. 6. Cap.

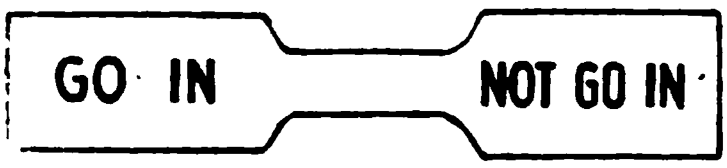
NOTE.—The above arrangements have been devised to obviate the risks of unequal expansion when heating up.

is done in different works in different ways, one method being to have the casing or rotor grooved first, the rotor put into place and grooves marked off relative to the grooves in part which has already been grooved.

Another method is the use of length sticks marked off from the drawing, the position of grooves, &c., relative to the shoulder on bearing and position of steam glands.

A similar length stick is also prepared for the casing grooves, an by applying the two length sticks together the position of rotor an casing blades can be ascertained with accuracy.

Turning of Rotors.—The rotor being marked off, gauges shoul be prepared for the width of groove and depth of groove. It is pr ferable to have two gauges for the width of groove, one to "go in and one "not to go in." The "not go in" gauge should be .003 t



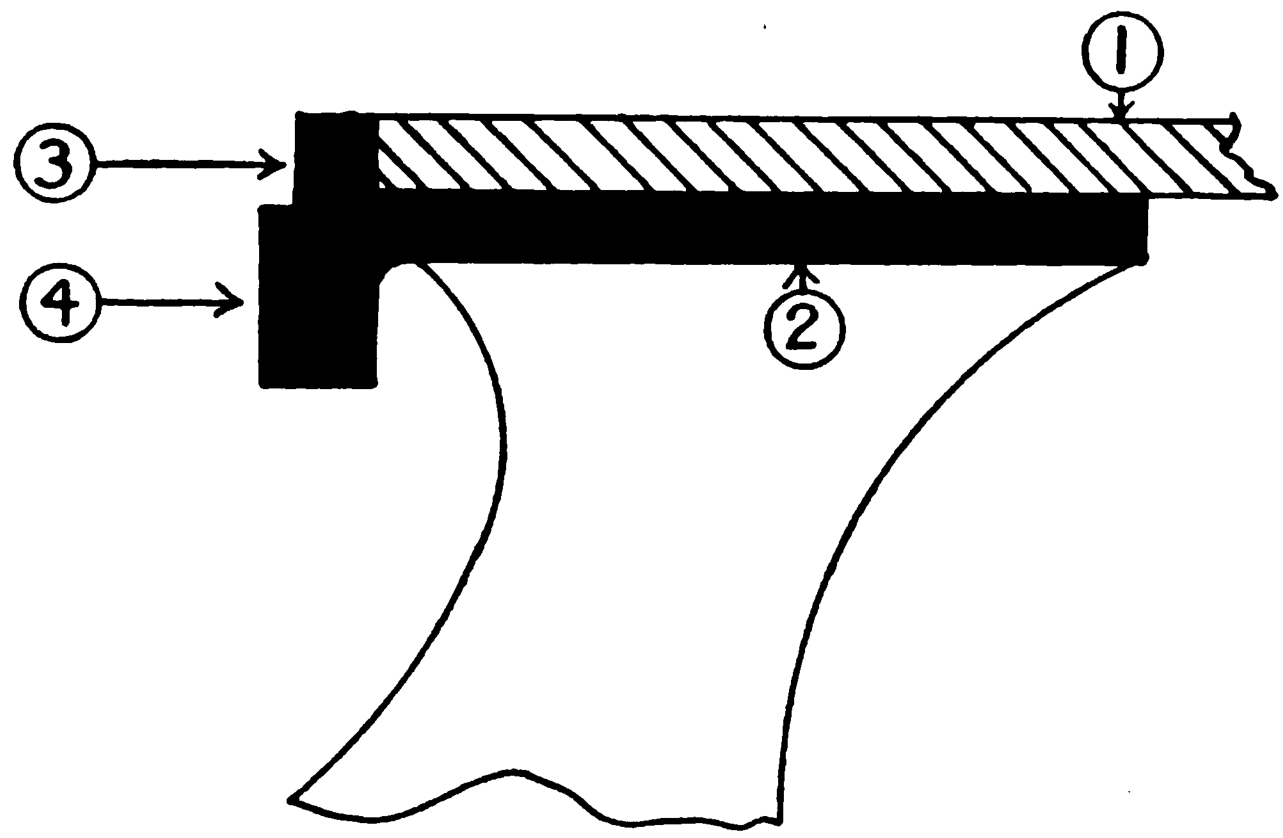
Test Gauge for Width of Grooves.

.004 larger than "go in" gauge. This ensures the groove being proper width to suit the blading. A complete set of gauges made f blading are supplied (standardised), the terms of which are follows:—

BLADING.

130 B, 140 B, 240 B, 250 B, 251 B, 260 B, 261 B, 271 B, 272 B, 280 281 B.

Packing pieces are of similar number, and have the describi

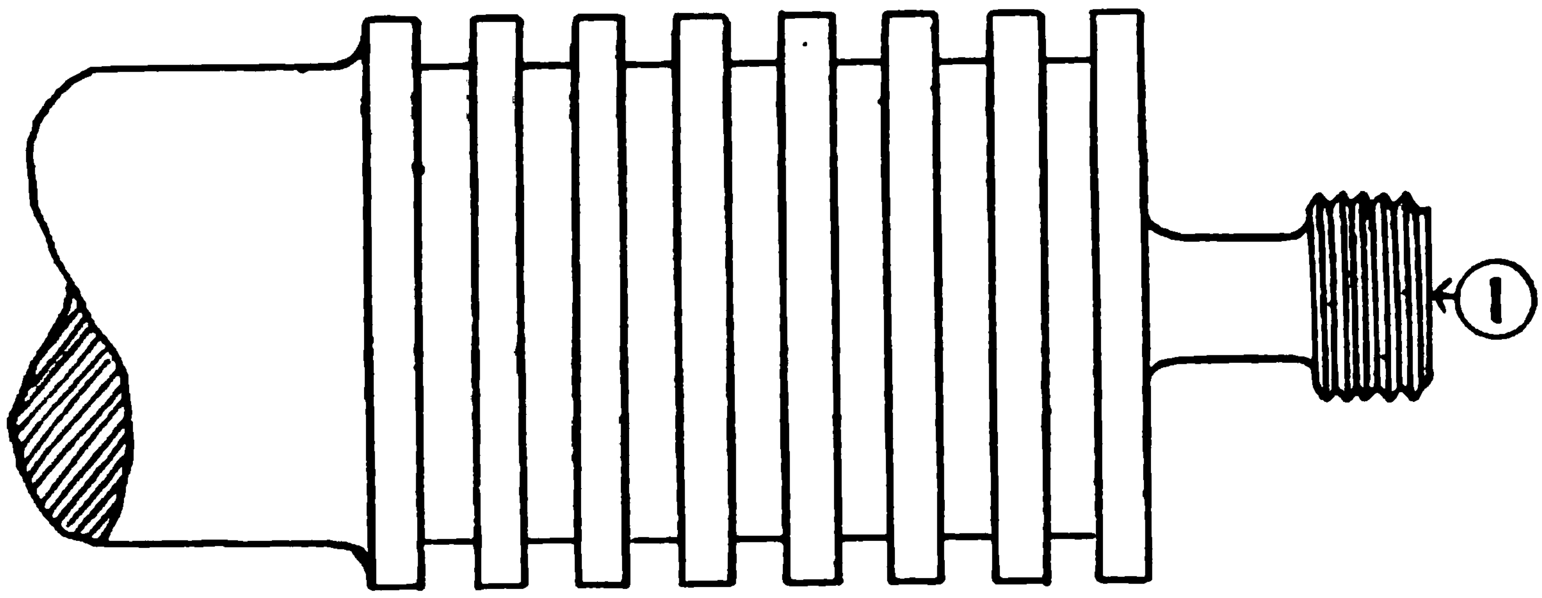


Wheel and Drum.

1. Rotor Drum. 2. Wheel. 3. Recess for Dummy Spigot. 4. Face for Dummy.

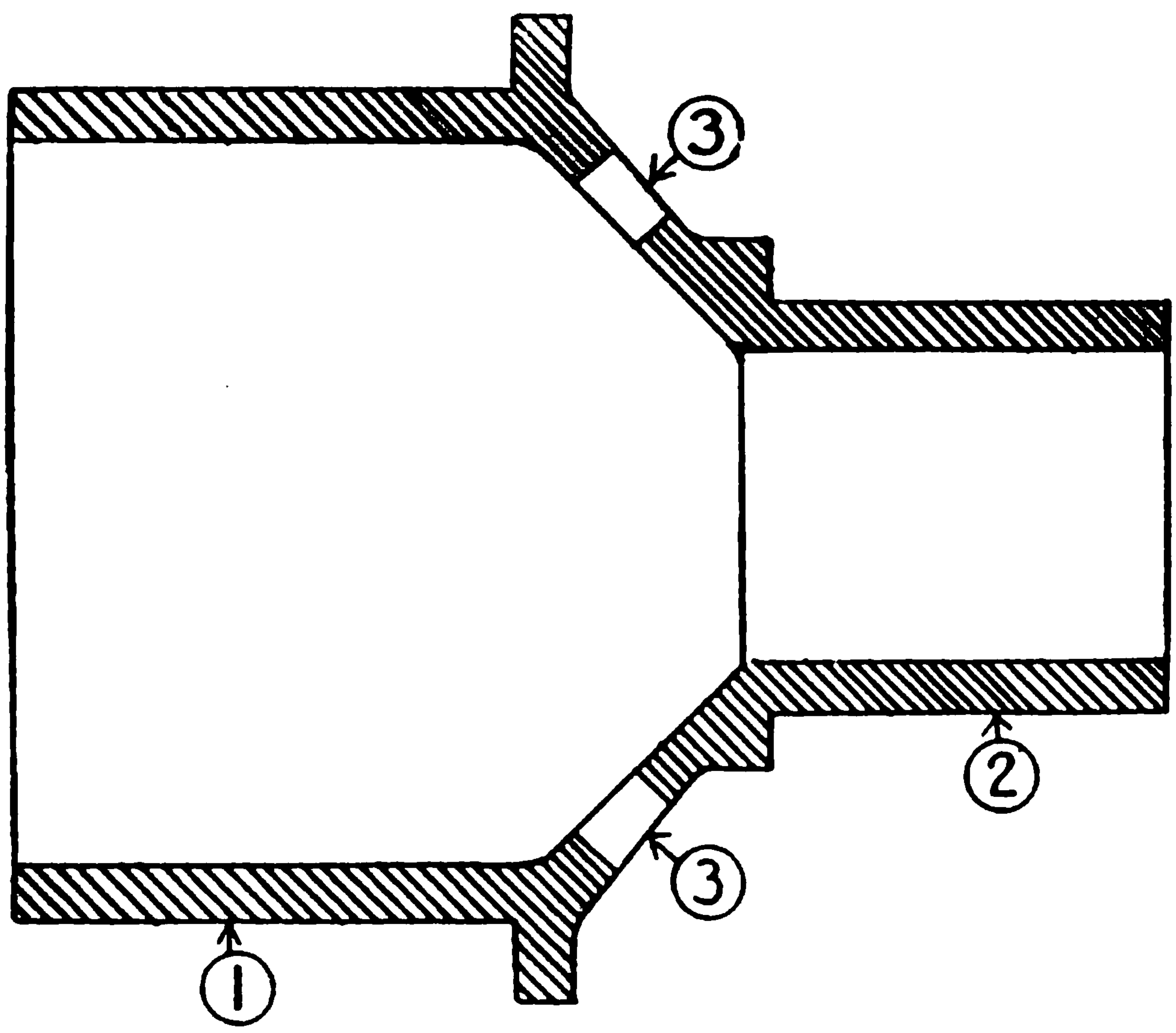
letter C and S after the number instead of B. B represents t blades, C the packing pieces for cylinder or casing, and S the packi pieces for rotor. The grooves are turned, and on each side of sa serrations are made, an indentation of V shape on each side of t groove. In shallow grooves the serrations are two in number, and deep grooves four serrations are formed on each side of groove. Th serrations are to assist in holding the blading in place, and will further described under the heading of "Blading." The serrating t

is usually made so as to have a cutting edge on each side; one side is cut, and then the tool is turned against the opposite side of groove and serrations formed. After grooving, the forward end of the rotor,



Thrust and Counter Gear (on Forward End of Spindles).
1. Worm connecting to Counter Gear.

that is the face of the forward wheel, is turned up to receive the dummy, which is bolted to this face. This face is spigoted to fit the dummy recess, thus ensuring the one being central to the other. In

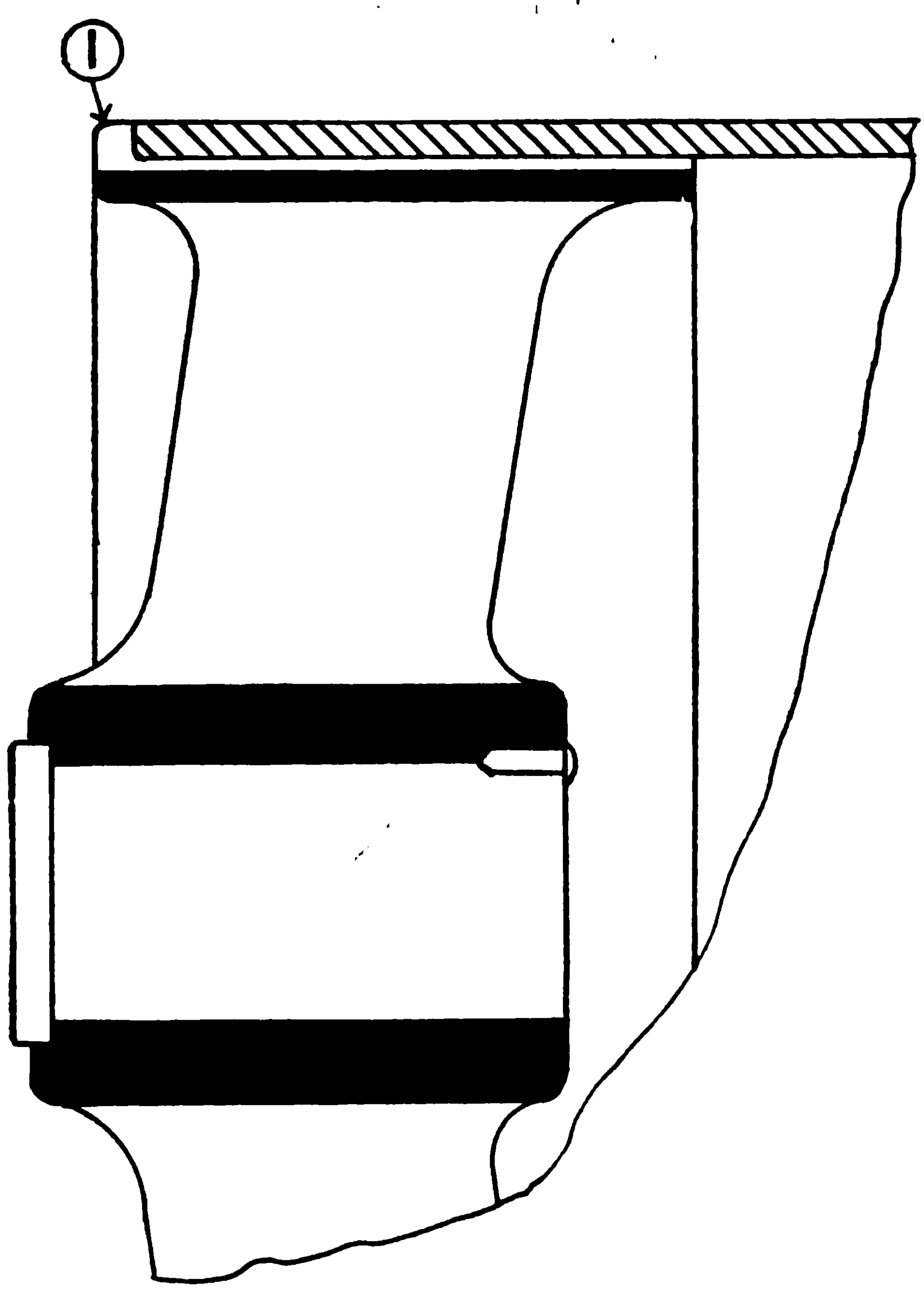


Junction Wheel (for connecting up the L.P. Ahead and Reverse Rotor Drums).
1. L.P. End of Wheel. 2. Reverse End of Wheel. 3. Holes in Wheel.

some cases the dummy is part of the wheel, but it is preferable to have a separate part, as it may be required to renew same, owing to pitting taking place and destroying the grooves in the dummy.

The bearing spindles are now turned up and finished. The part

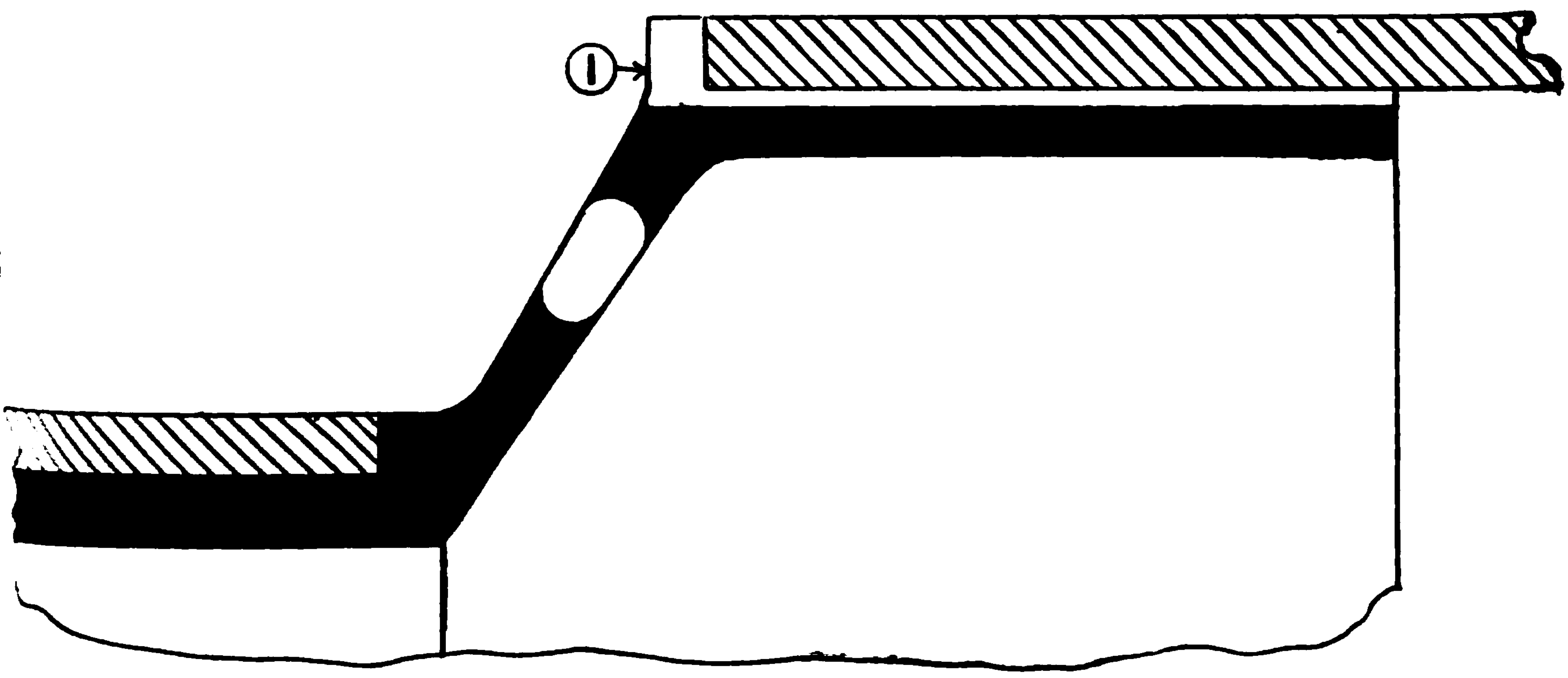
of the spindle nearest to the wheel is of larger diameter than that forming the bearings, and this part is grooved for the steam gland packing. These grooves are usually $\frac{3}{16}$ in. deep × $\frac{1}{8}$ in. broad, and into these brass strips are driven, thus forming what is termed "labyrinth" packing. This is described more fully under the heading of "Steam Glands." At the end of the part which forms the bearings



Wheel.
1. Drain from Inside of Drum.

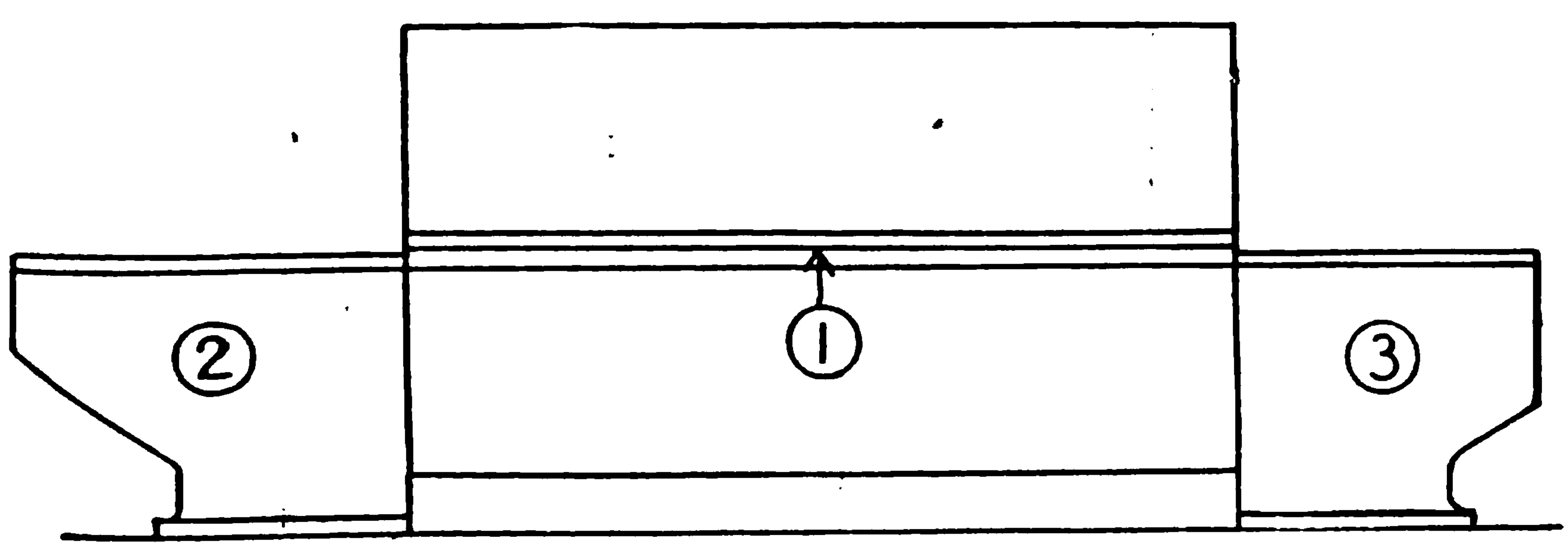
a series of V indentations are made; these act as oil deflectors, and prevent the supply of oil which is forced through the bearings from escaping along the shaft. A low-pressure rotor differs only from a high-pressure rotor in that in average size turbines the rotor consists of one large drum and one small drum. The small one is for reversing purposes, and is usually at the after end of the low-pressure rotor. The rotor is built in the same way as an H.P. rotor, and has a con-

necting wheel between the large drum and the small one; this wheel is termed a "junction wheel." The wheel is turned all over, and is of two diameters, one to go into the aft end of the L.P. drum and the other to go into the forward end of astern or reverse drum. Through the wheel holes are bored, which allow any steam which may get into



Junction Wheel (for connecting up the L.P. Ahead and Astern Drums).
1. Drain from Inside of Drum.

the inside of the rotor to escape to the exhaust. It will be seen that the interior of an L.P. rotor is thus in a vacuum. The rotor drum is pinned to the wheels in a similar manner to the H.P., and is turned up and grooved. The rotors are parallel, and the casing is stepped to suit the different lengths of blades of each expansion.

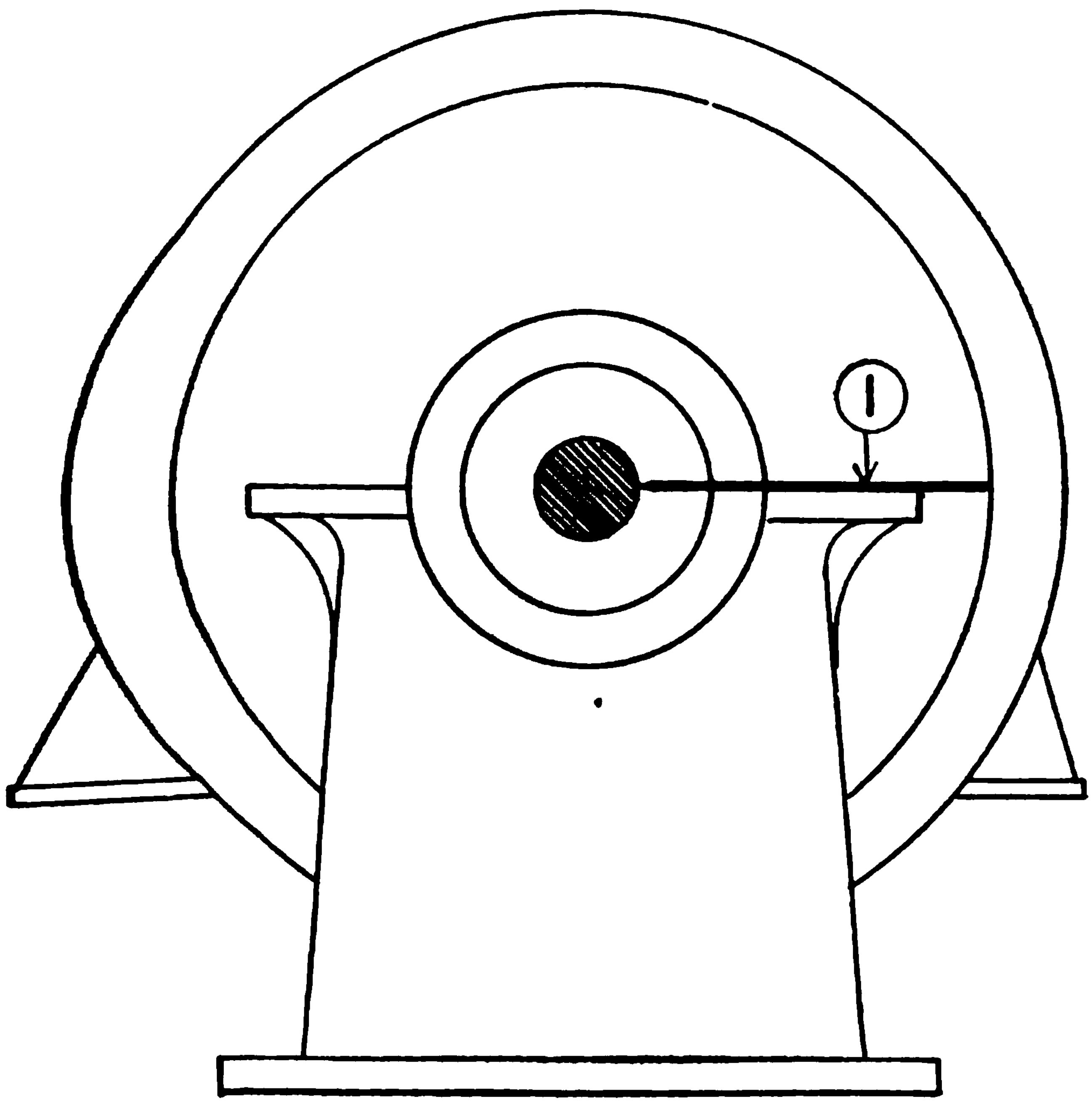


H.P. Turbine Casing.
1. Horizontal Joint. 2. Bearing Stool and Thrust. 3. Bearing Stool.

Turbine Casings.

Turbine casings or cylinders are of cast iron. There are several different designs in regard to the ribbing and strength of same. In merchant steamer work they usually have heavy bulb ribs running longitudinally and circumferentially about the shell. The high-

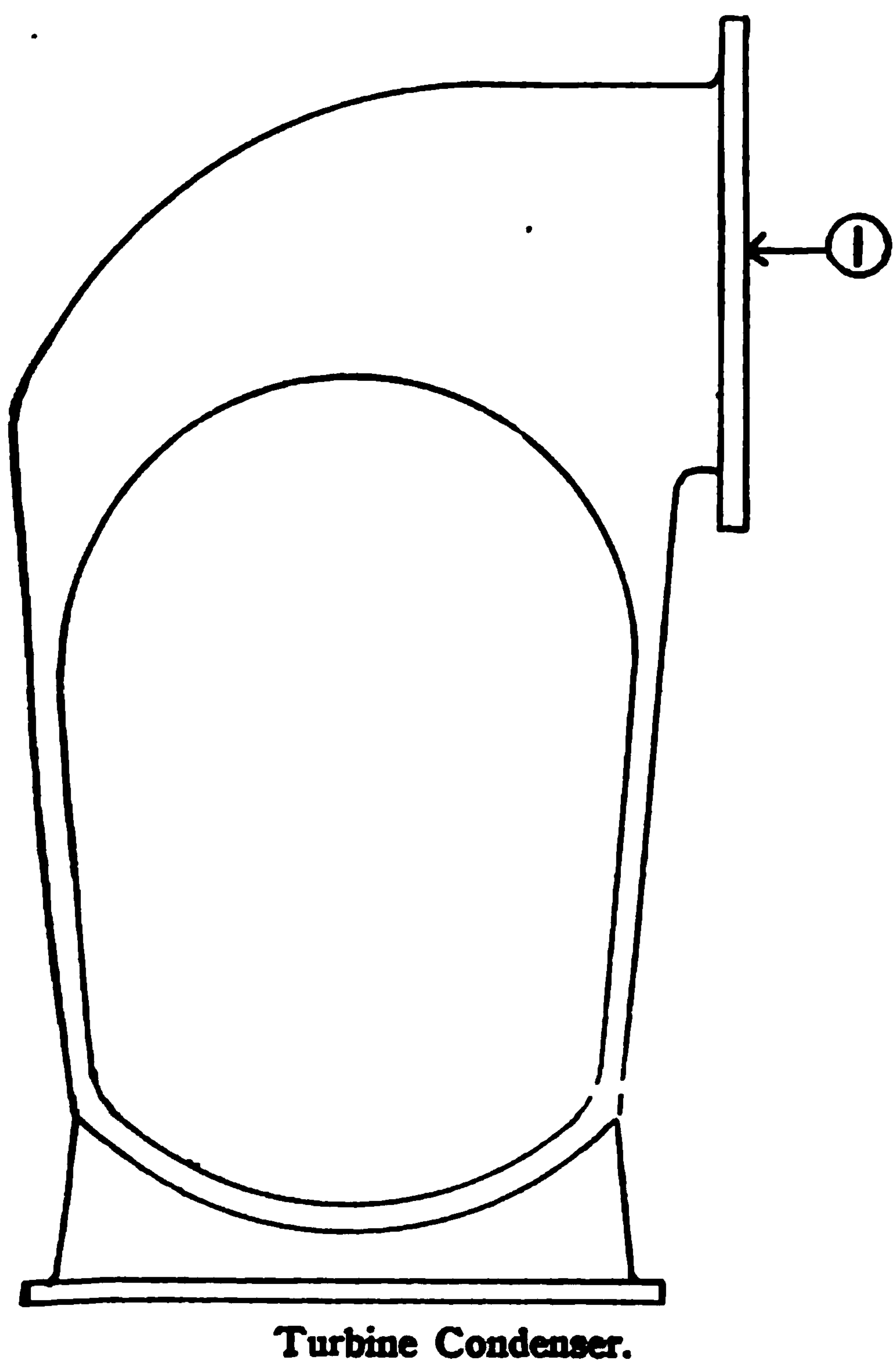
pressure casing in average size turbines usually consists of two parts, the top and bottom halves of casing. The low-pressure casing is usually in four pieces, the top half being made up of two parts, and the bottom half of two parts. These two parts are bolted together, forming a circumferential joint. In large turbines the casing sometimes consists of six or eight pieces—this is to ensure good castings. When casings are delivered from foundry to shop, they should be calipered all over to ascertain thickness of metal, and then put on surface table and drawn in. The horizontal joint is then planed in both halves, and marked off for boring bolt holes. In an L.P.



Gauging Bore of Casing.
1. Gauge.

casing, there being four parts, after the horizontal joint is machined and bored in one section, and four corner holes in the same are widened and temporary bolts fitted; the casing is then sent to the turbine-boring machine, and inside of casing rough bored to $\frac{1}{8}$ in. smaller than the finished size. The same operation takes place with the other section, after which the bolt holes in circumferential joint are bored. The high-pressure casing goes through the same operation of rough boring, and is now ready for water testing. This casing is usually tested to from 230 to 250 lbs. per sq. in. The forward end of the low-pressure casing is tested to 50 lbs. per sq. in., and the aft end to 30 lbs. per sq. in.

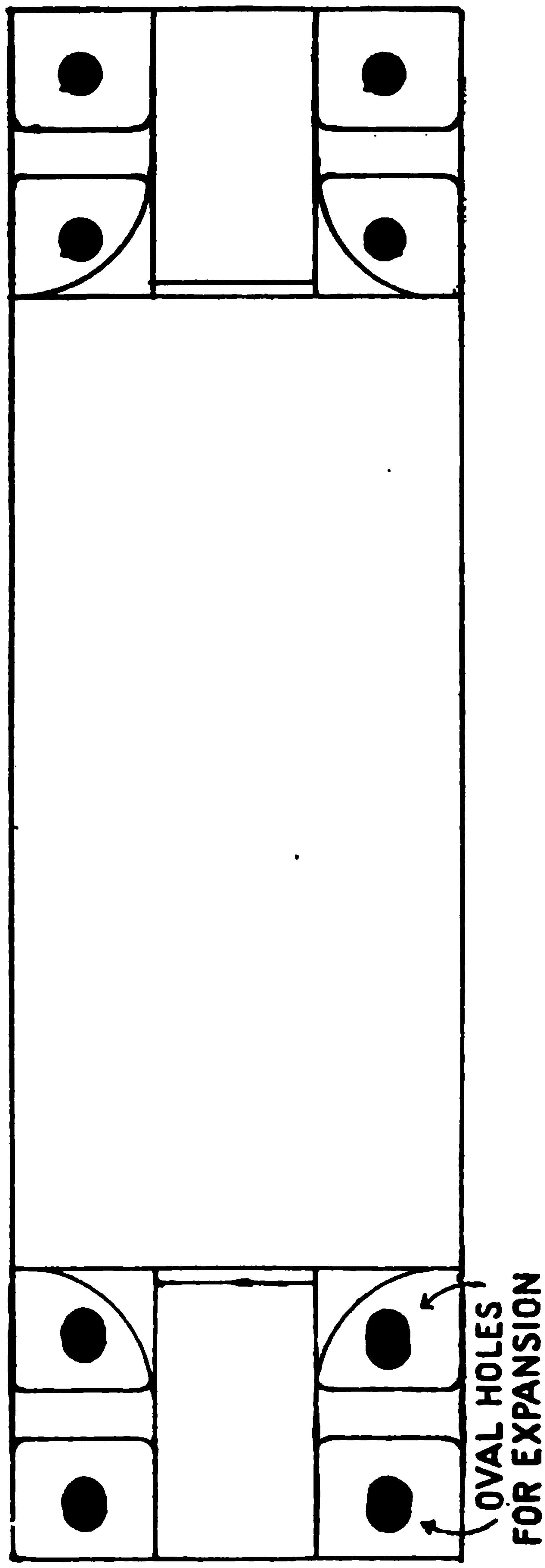
After the casings are water tested, they are "steamed" for twelve to twenty-four hours at a pressure of from 20 to 50 lbs. This is of great importance, as it tends to thoroughly expand the metal, and if not properly carried out, may result in the distortion of the casing when running the turbines, and cause destruction of blades. Each section of the low-pressure casing being separately tested, the two halves are bolted together, and steamed in a similar manner. After steaming, the joints of the casings are bedded, and in the case of the low-pressure casing, the circumferential joint is made. This



Turbine Condenser.
1. Exhaust from L.P. Turbine.

is usually made of mastic cement or red lead put on in the consistency of thick paint. The casings are now ready for final boring and grooving. In setting up in the machine, care should be exercised to ensure casing joint being exactly central to boring bar.

The casings are bored out to gauges applied from boring bar, and are marked off for grooving. The ends of the casing where the dummy is bolted to, and also the face where the astern cylinder is bolted to, in low-pressure turbines, are turned up. Usually in average size turbines the main bearing stools are part of the bottom



Plan of Casing showing Oval Bolt Holes at Forward End for Expansion.
The after end is bolted rigidly to the seating of the steamer.

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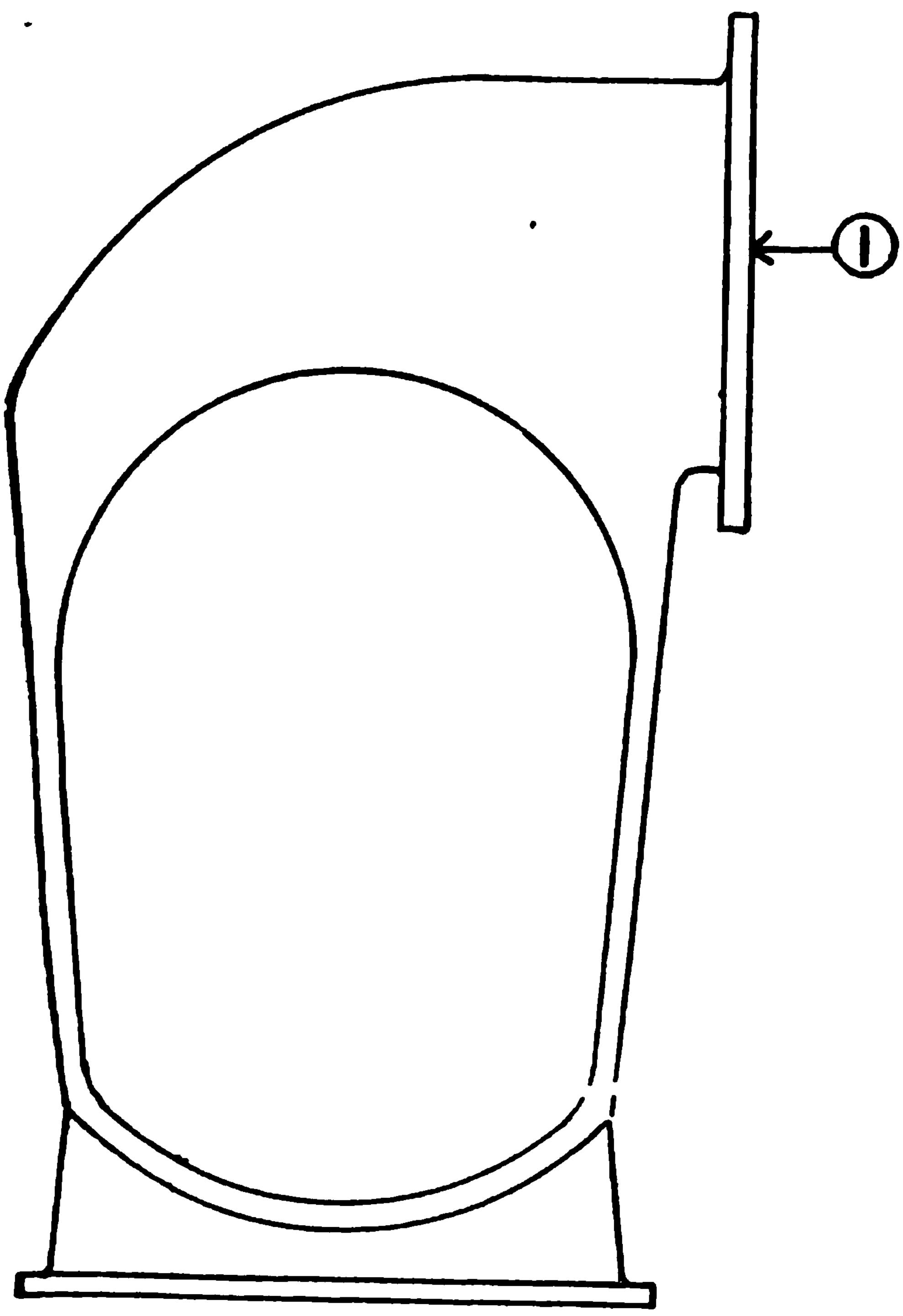

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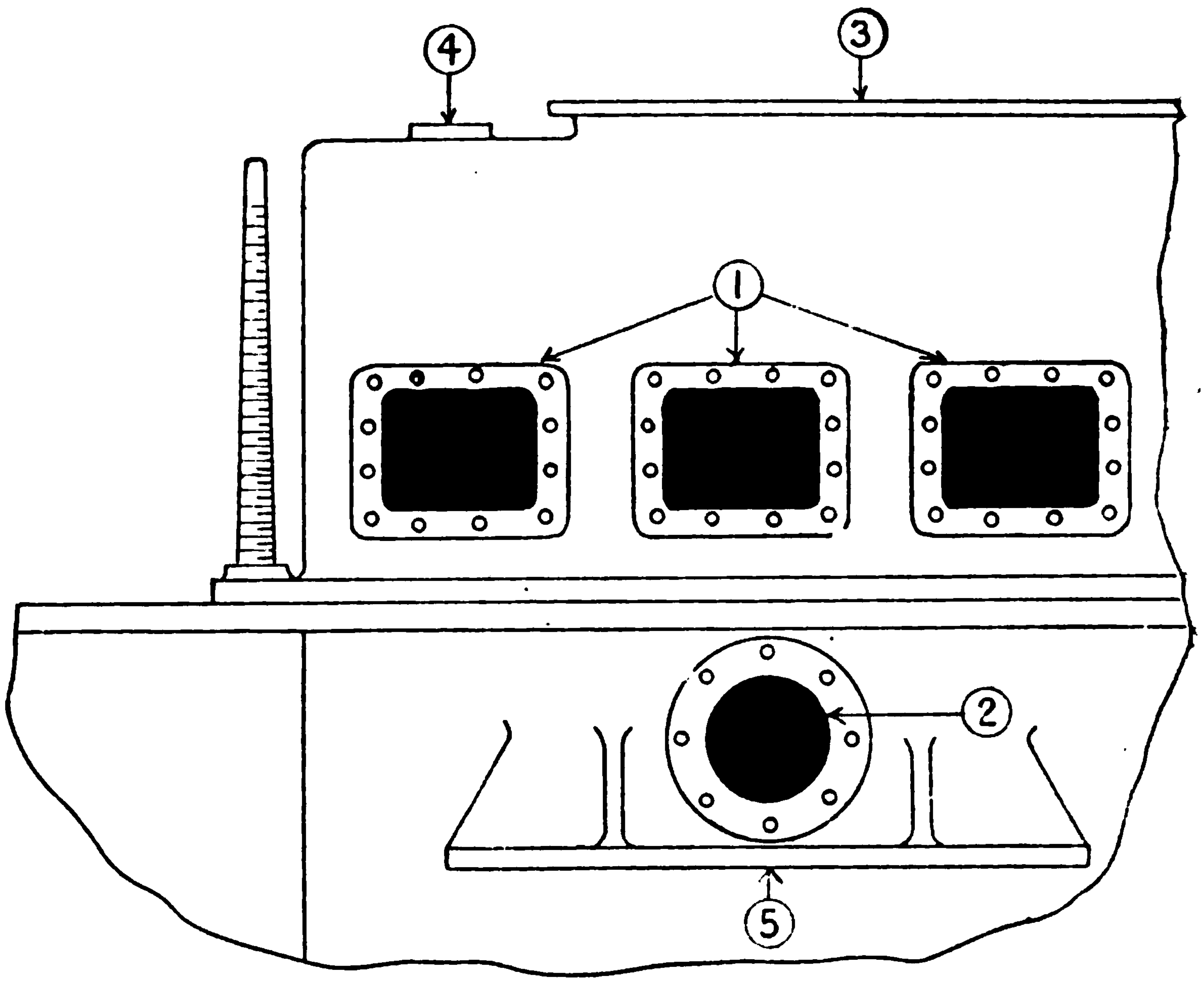
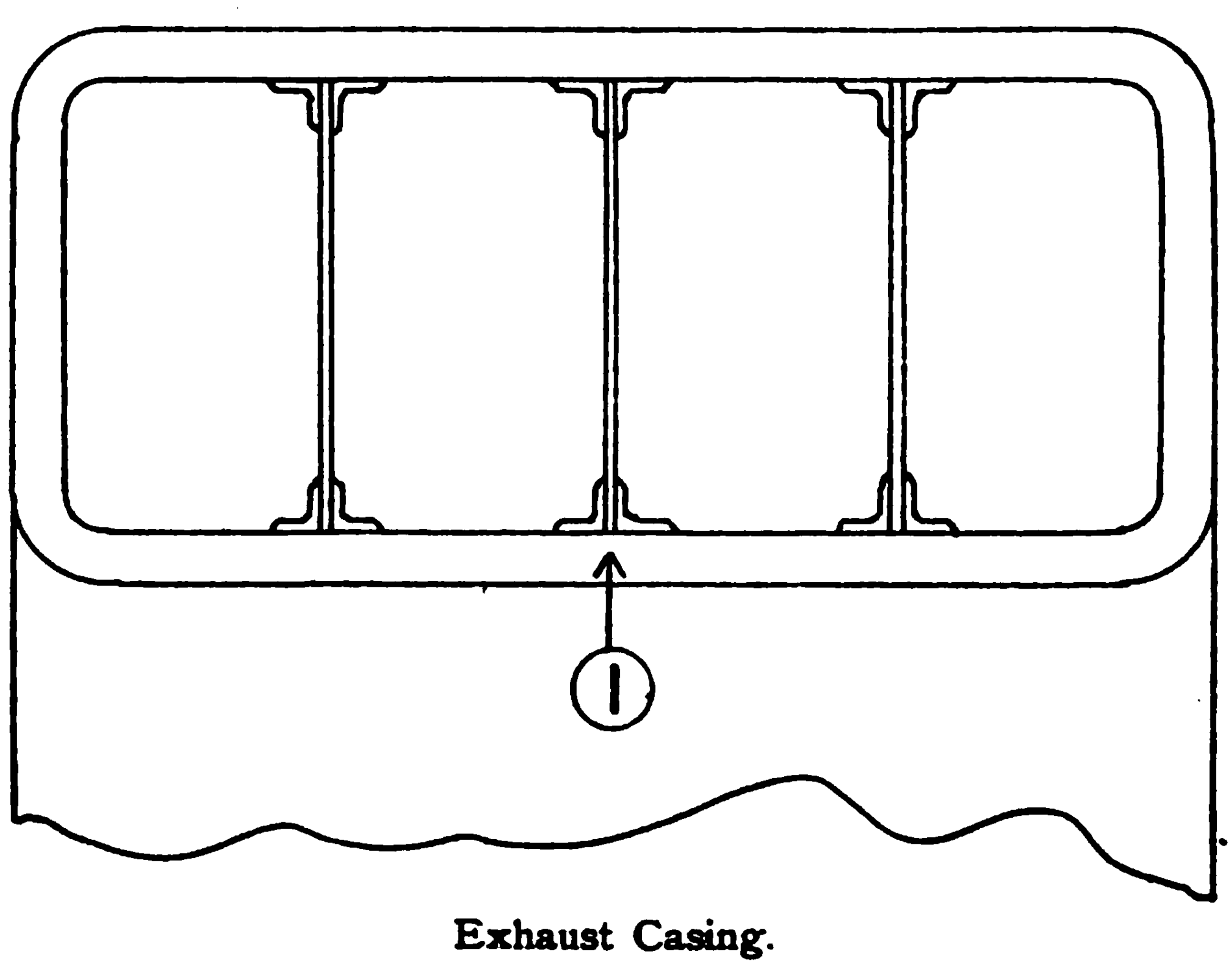


After End of L.P. Casing.

1. Access Doors to astern casing.
2. Access Door to bolts of astern cylinder feet.
3. Exhaust to Condenser.
4. Relief Valve.



Exhaust Casing.

1. Plate and Angle Iron Stays.

Views of H.P. and L.P. Turbines.

Showing Ahead H.P. Dummy, and the four H.P. Expansions, also the L.P. Ahead and Reverse Dummies, and the eight L.P. Ahead Expansions, and four Reverse Expansions.

NOTE.—Each H.P. Expansion contains sixteen rows of blades, and each L.P. Expansion contains eight rows of blades.

[*To face page* 80.

View of L.P. Turbine and Reverse Turbine in Lower Half Casing.

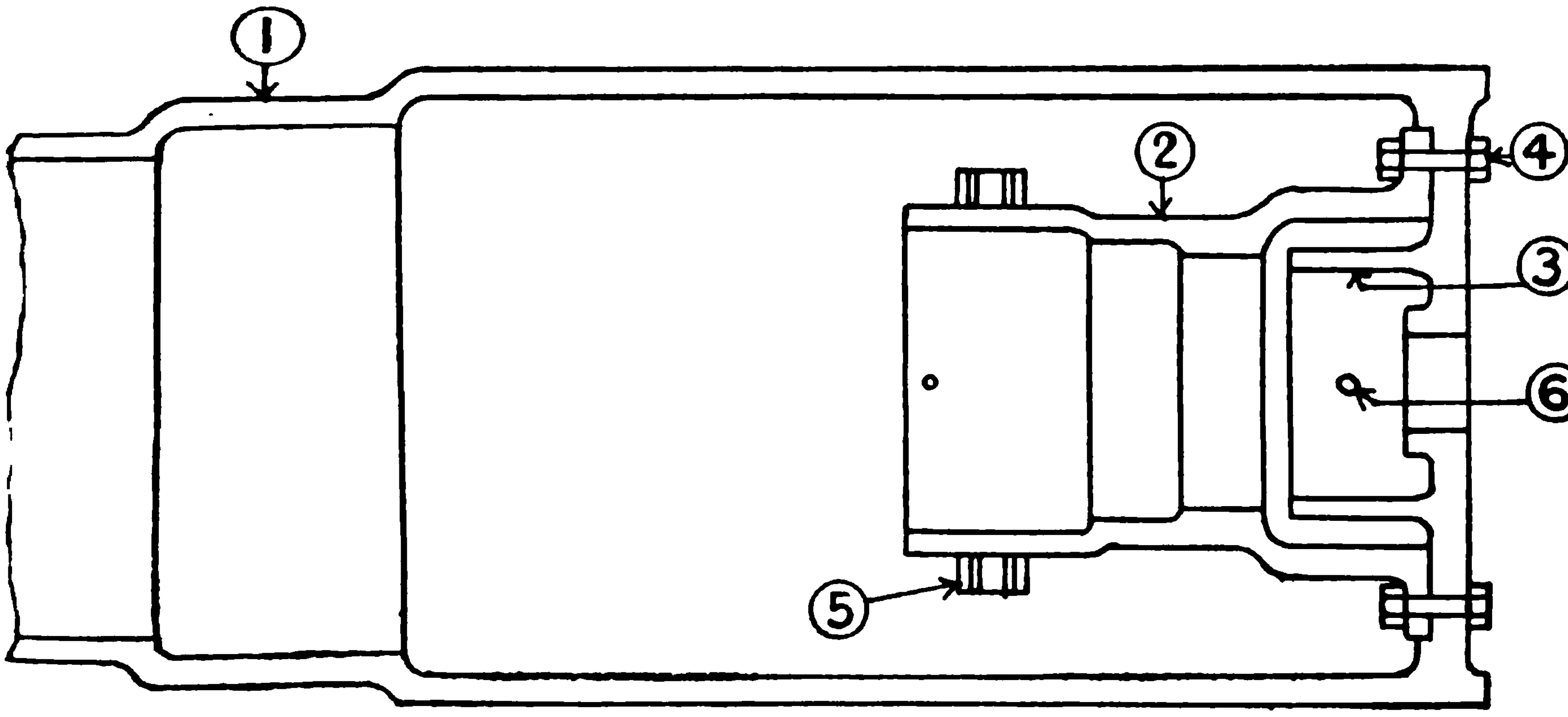
Showing (left) "Wheel" or "Centre" in Lower Half Casing, Ahead Dummy Piston, Ahead Expansions (eight in number), Reverse Expansions (four in number), and Reverse Dummy Piston.

NOTE.—The last three Ahead Expansions have blades of equal height, but of wider spacing, and the last two Reverse Expansions have blades of equal height, but of wider spacing. The steam pipe shown is only that fitted in the shop for "steaming" purposes.

"The Marine Steam Turbine."

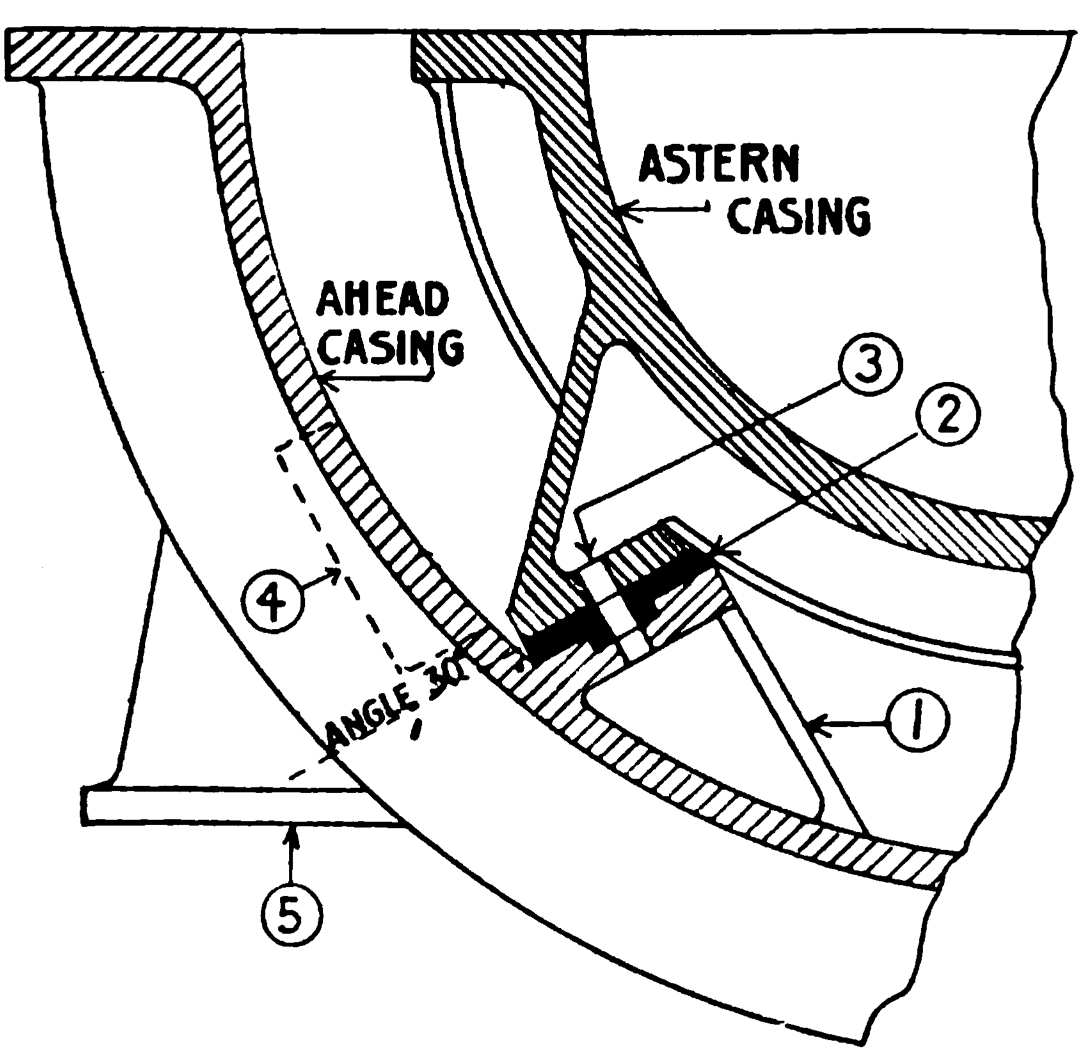
[*To face page* 80.

half of casing, and in this case they are bored out in this machine, thus ensuring the bearings being concentric to the bore of the casing.



L.P. Casing and Reverse Casing.

1. L.P. Ahead Casing. 2. Reverse Casing. 3. Reverse Dummy. 4. Bolts of Reverse Casing. 5. Feet of Reverse Casing. 6. Dummy Casing Drain.



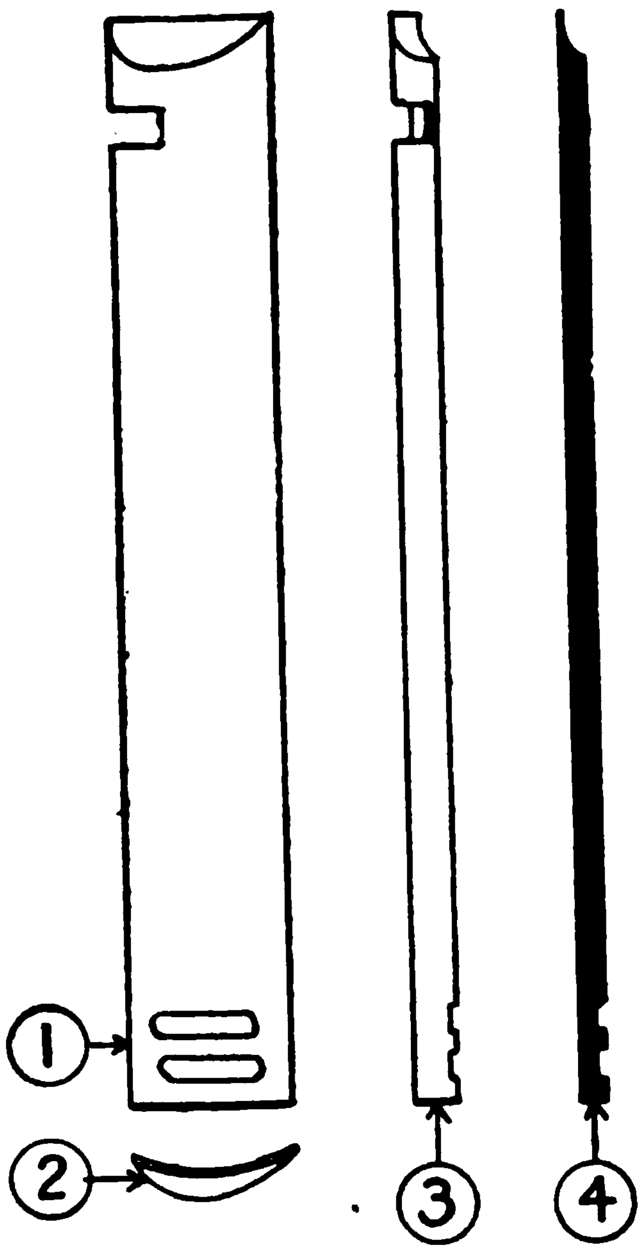
Astern Turbine Casing.

1. Bracket cast on L.P. Ahead Casing to receive reverse casing feet. 2. Brass Chock. 3. Hole ⅛ in. larger than Bolt to allow of expansion. 4. Access Door. 5. Feet of L.P. casing.

For turbines up to 18,000 horse-power the usual arrangement is to have the astern casing or cylinder inside the after end of the low-pressure. The astern casing is bolted to a flange inside the low-

pressure casing, and is also supported by diagonal feet on each side which sit on a ledge on the wall of the low-pressure casing. A brass chock is fitted under this foot, and a bolt put in, a loose fit, so as to allow the astern cylinder to remain central when expansion comes on after casing is heated up. The astern cylinder is turned and grooved in a similar manner, and is water tested to a pressure of from 100 to 150 lbs. per sq. in.

Blading.—The blading for Parsons' turbine is of brass, and is manufactured by various firms: it is usually delivered in lengths of from 5 to 6 ft. The turbine proper being formed of blades of



Blade with Thinned Tip.

1. Elevation of Blade.
2. Plan of Blade.
3. Edge view of Blade.
4. Section of Blade.

various lengths and spacing, which are termed "expansions," the blades are cut to lengths in a machine which shears them off and at the same time stamps a double or treble groove on the end.

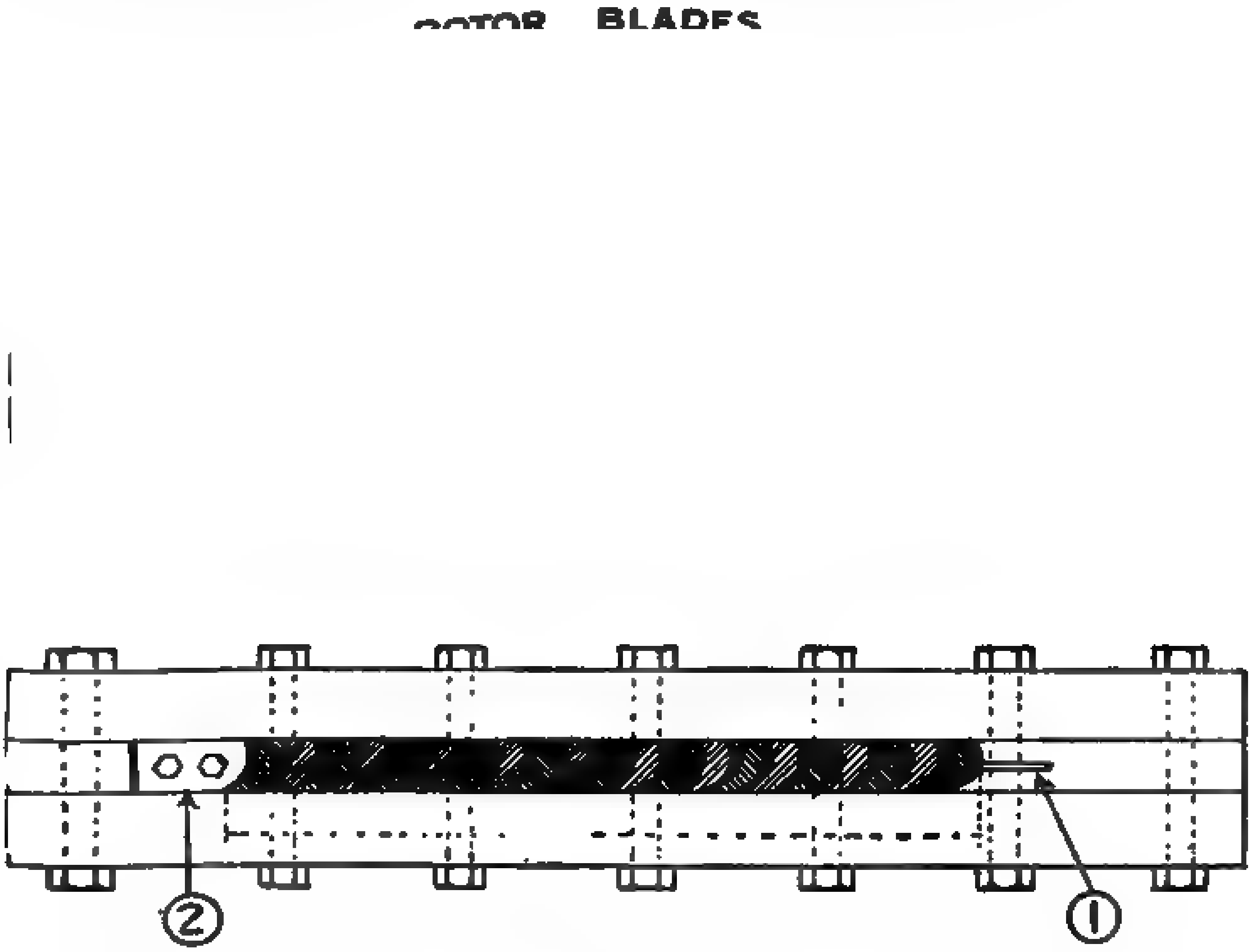
These serrations are to assist in holding the blade in place (see sketches). The blades being of various lengths, the rotor is left parallel and the casing stepped to suit this. After the blades are cut to length they pass to a machine which cuts a groove about $\frac{1}{4}$ in. from the end of blade remote from the serration, and into this groove the binding wire is fitted. From this machine the blades pass to a machine which thins down the point on tip of blade to about .001 in.

Rotor Partly Bladed.

(33 Knot Destroyer.)

To face page 82.

so that in the event of blades coming in contact with the casing or rotor they will simply turn over or wear down without serious damage resulting. There are several different systems of blading, the original system being to blade the rotors and casings by hand, but this has been superseded to some extent by what is termed "segment" type blading. There are two types of segment blading, "Parsons'" and "Willans & Robinson's." The Parsons segment blading is built up in

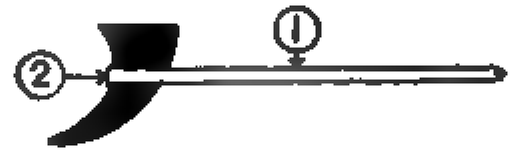


"Segment" Block for Blading.

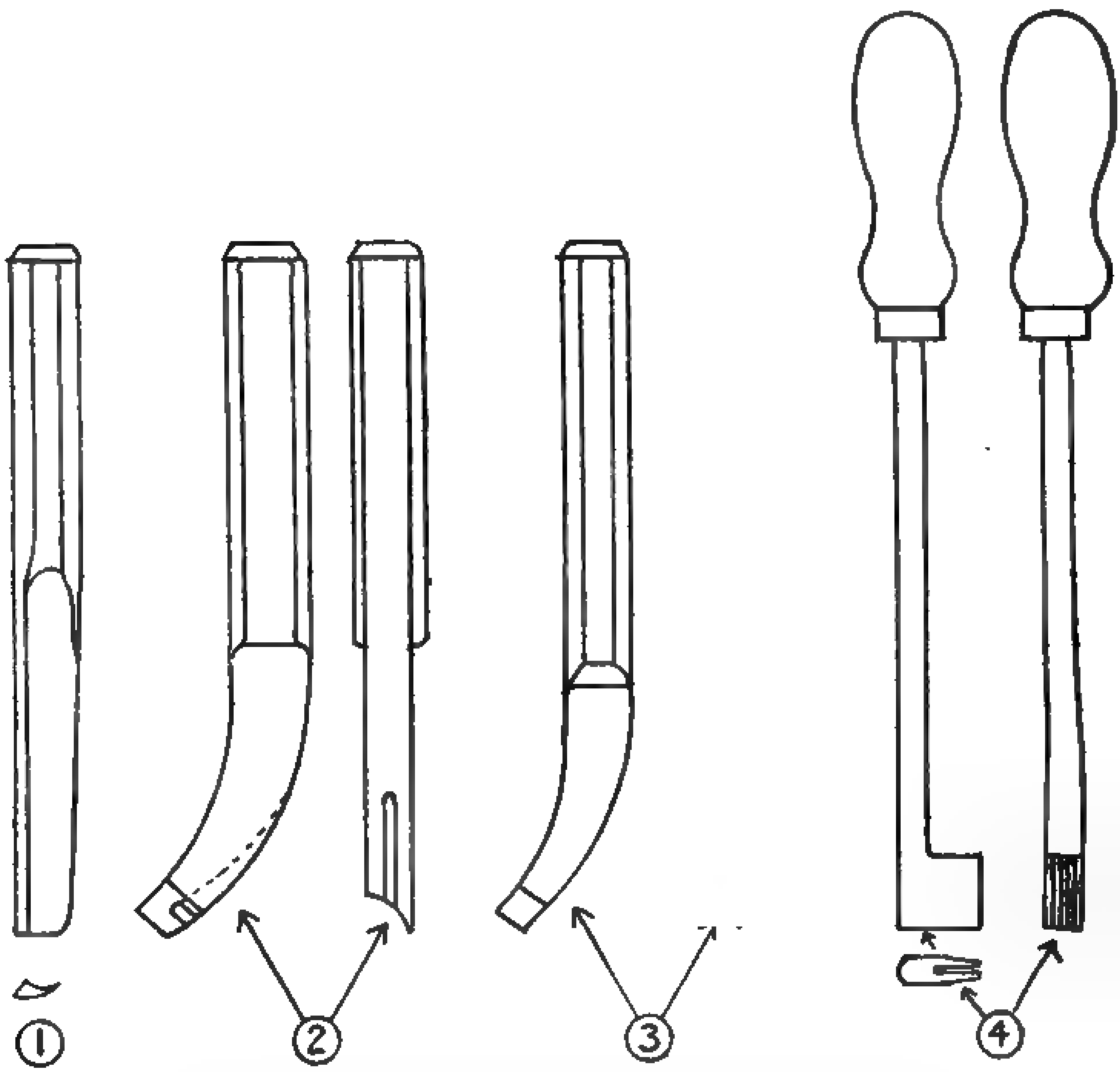
1. Brass Wire through Blades and Packing Pieces. 2. Stop Piece pinned down in Groove. 3. Brass Binding Wire lashed and soldered to Blades.

two cast-iron plates which are bolted together, and have a groove, equivalent to the groove in casing or rotor for each expansion, turned in them. Usually one side of the block is for rotor and the other side for casing, the block being cast concave on one side and convex on the other. The blades have an $\frac{1}{8}$-in. hole drilled through them at the root, through which an $\frac{1}{8}$-in. brass wire passes. Between each blade there is a packing piece section. The packing piece is of soft brass and is delivered in lengths, which are cut up in revolving circular saws, in sections equal to the depth of the groove into which they are to be fitted. It is preferable to leave packing pieces

$\frac{1}{32}$ in. deeper than groove, so that when caulked into place they will then be flush. Through these packing pieces an $\frac{1}{8}$-in. hole is also drilled at



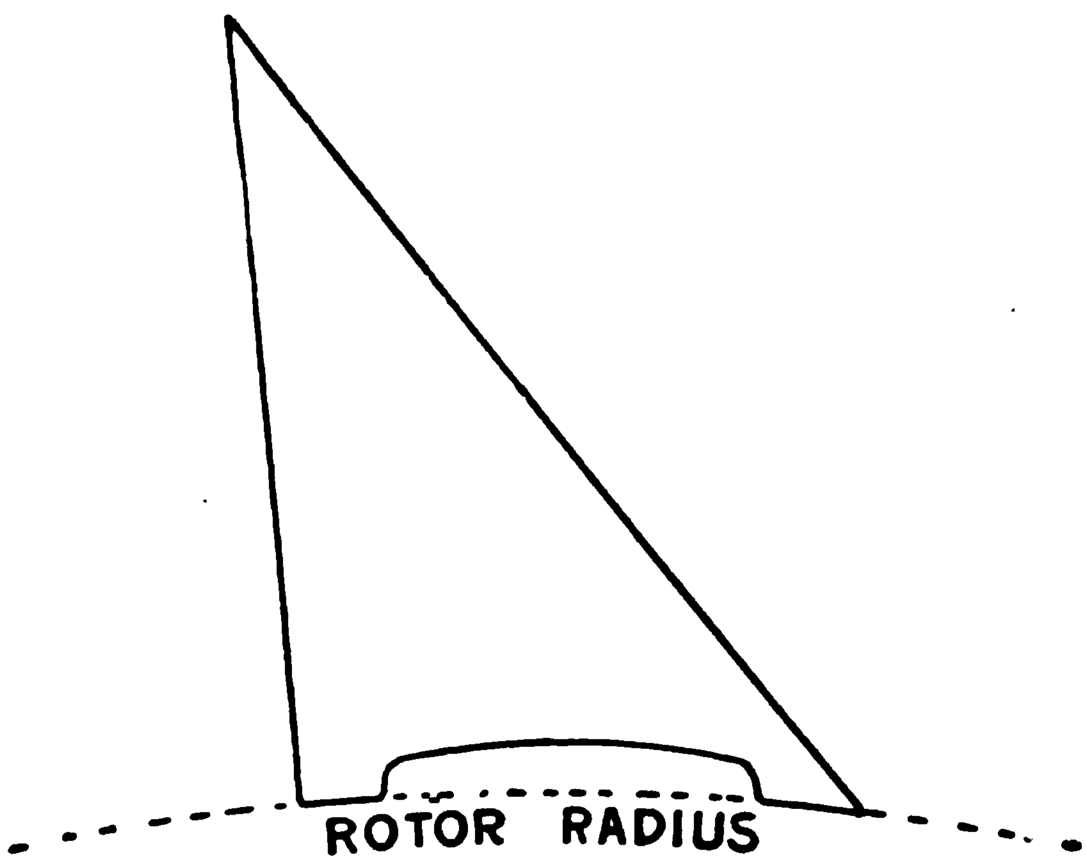
Segment Blading
(First Operation.)
1. Brass Wire $\frac{1}{8}$ in. diameter passed through Blades and Packing Pieces.
2. Brass Wire riveted over Packing Piece.



1. Caulking Tool for Blades. 2. Driving-up Tool for "Segment" Blading.
3. Driving-up Tool for Standard Blading.
4. Tool for "Setting" Blades (sometimes called a "Twister").

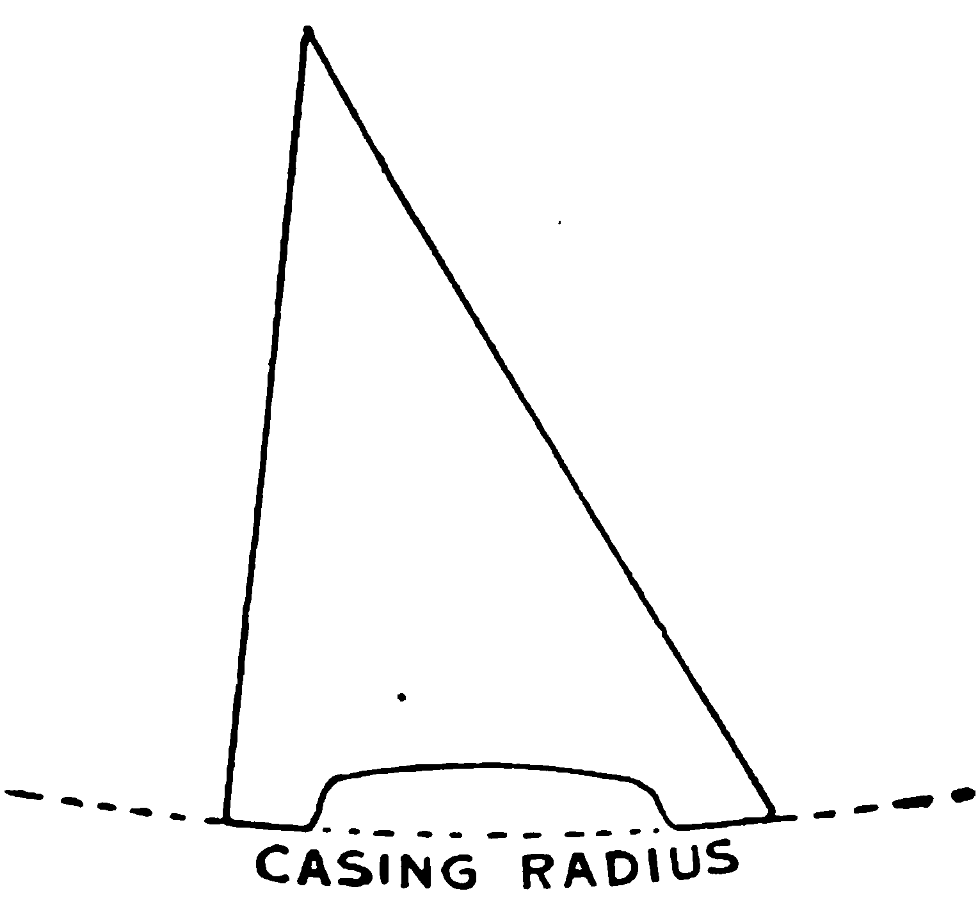
No. 1 is employed in caulking up the packing pieces and blades. Nos. 2 and 3 are employed in driving up the blades and packing pieces endways previous to caulking. No. 4 is employed in twisting or setting the blades to the correct angle and position after caulking.

the requisite angle in a machine, in which are two dies of the same shape as the back and front of the packing piece. These catch the packing piece and hold it at the proper angle, so that the hole is in



Set-Square for Setting Rotor Blades.

line with the hole in the blade. After being bored the packing pieces are annealed by heating to a dull red heat and allowed to cool out. A packing piece is taken, and through it a length of $\frac{1}{8}$-in. brass wire, a



Set-Square for Setting Casing Blades.

little longer than the segment required, is passed and the end of the wire riveted over; this is put into the groove in the building block and driven up against a stopper pinned into grooves. A blade is then

passed on to wire, next a packing piece, then a blade, and so on. About four blades and five packing pieces are put on, and the lot driven up against the stopper. This is done by a caulking tool having a groove cut on its under side so as to clear the wire which passes through the blades and packing pieces (see sketch). This operation is carried on, care being taken to see that all blades and packing pieces are resting on bottom of groove in building block until the required length of segment is built. At the end of segment two or three packing pieces are put in and driven hard up against last blade so as to ensure segment remaining firm until it is completed. A segment starts with a packing piece and ends with a blade. The

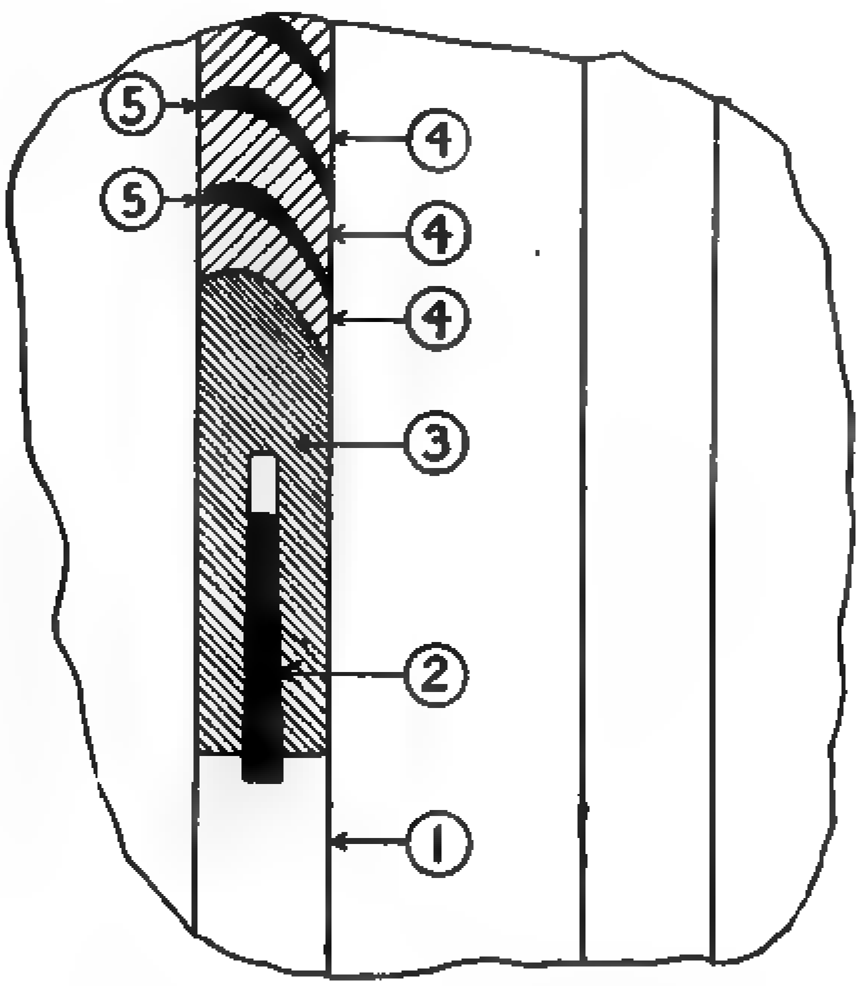
Caulking Tool for "Segment" Blading.

"Segment" Blading.

View of Blade caulked into groove with $\frac{1}{8}$-in. brass wire passing through Blades and Packing Pieces.

blades are now trued up or spaced. This is done by using a set-square suitable to the radius of the casing or rotor for which the segment is intended. After trueing up the blades the binding wire is fitted into groove at top of blades, and in blades of over 3 in. in height this wire is usually laced with copper wire. The wire is now soldered with silver solder, using a gas blow lamp and a flux of fine powdered borax. In segments of long blades there is usually a middle binding wire which is soldered into place, but is not laced. After the segment is completed the packing pieces at the end are taken out, and the wire is cut and riveted over the hole at the root of the blade. One segment for each row is usually left with two or three blades unsoldered so that segment may be made the proper length for closing row. The

original Parsons' method of blading in the case of a rotor is as follows. Into the groove a stop piece is put, this stop piece being held in place



Method of Blading Rotors.

1. Groove in Rotor Drum.
2. Steel Wedge to Lock "Stop."
3. "Stop," against which the first Packing Piece is Butted.
4. Packing Pieces.
5. Blades.

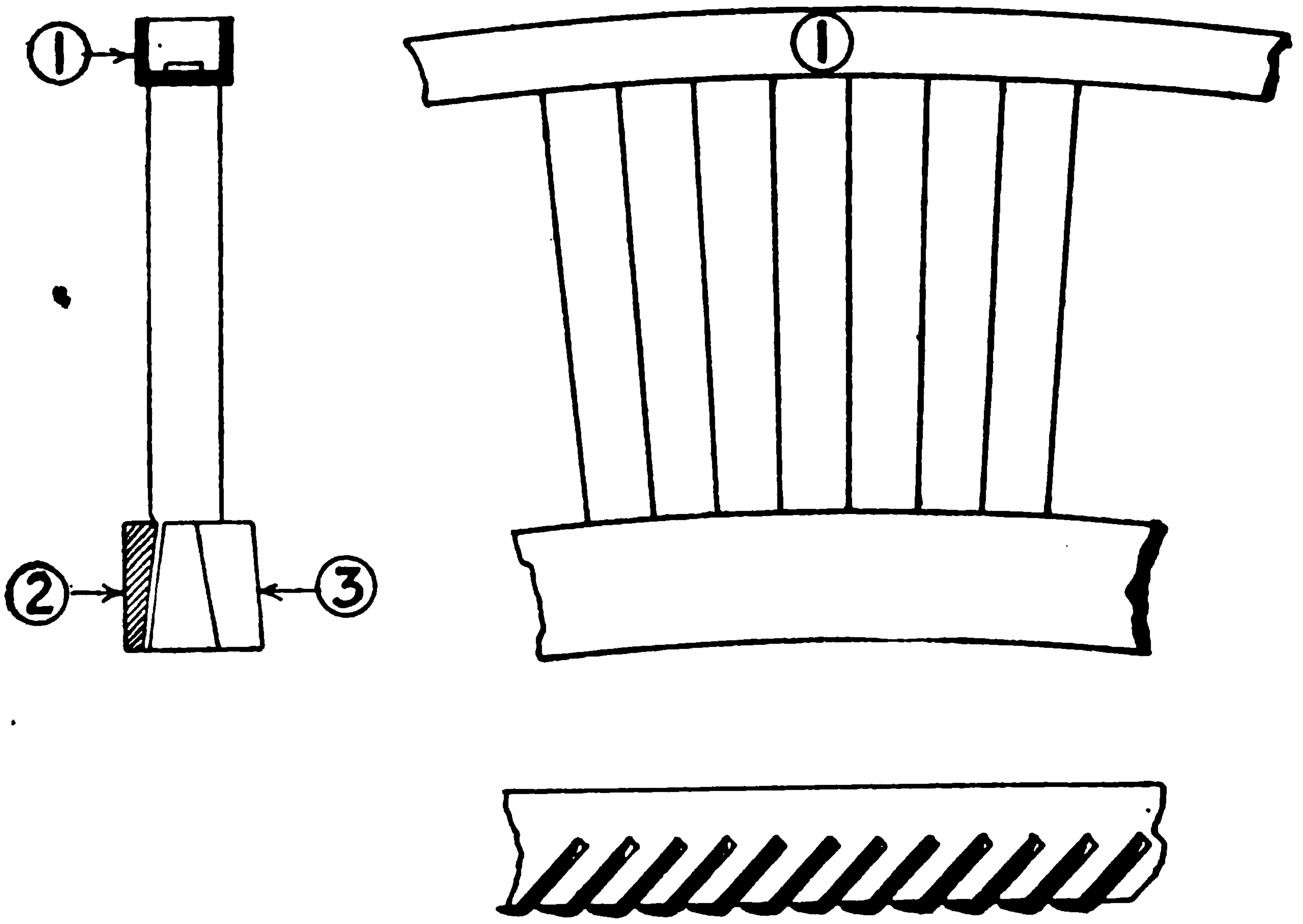
by having a saw-cut in one end of it into which a steel wedge is driven, thus binding it against the sides of the groove. The blades and pack-

Groove and Serrations for Extra Long L.P. Blades.

Groove and Serrations for Short and Medium Blades.

ing pieces are driven up against this, four or so blades at a time, until the row is completed. It is then tuned and soldered in a similar

manner as in segment building. The Willans & Robinson segment building consists of a base piece or foundation ring, having saw-cuts on an angle equal to the angle of the blade of which segment was to be built. The blades are cut to length in the usual way, and then stamped on the bottom to the shape of an L. The straight side of the blade is put into the base ring with the tail lying against the side of ring, and the whole put into a building block, which is screwed up against blades and rings, thus binding same. The blades are tuned and soldered in a similar manner as in Parsons' blading. Over the tips of the blades a "shroud" is fitted of channel shape ⊔. This shroud is of brass about $\frac{10}{1000}$ thick, and in the event of blades coming



Willans & Robinson System of Blading.

1. Channel Ring Shroud. 2. Caulking Ring.
3. Tapered Packing Section Ring.

in contact would wear and cause no serious damage to blading. The grooves in the casing or rotor for Willans & Robinson's blading are turned wider than the width of the foundation ring. Into the space left after segment is put in, a soft brass packing strip is put and is caulked into place by air hammers. The sides of the groove are dovetailed to an angle of from 4° to 6°, and after packing strip is hard caulked the whole segment is firmly held in place.

Blading of Rotors and Casings.

Rotors.—The segments are fitted into rotors by having a stop piece, as before described, put into grooves. Against this stop piece

the first segment is put, and driven up against stop; the other segments are then put in and row completed, the stop piece being taken out before putting in last segment. Two or three packing pieces in each segment are slightly caulked with a vertical caulking tool so as to prevent them from rising up or falling out; the row is completed by the last segment having three or four loose blades at the closing end; if required, one or two are taken off until the requisite length is obtained. The blades are now caulked in place, which operation is

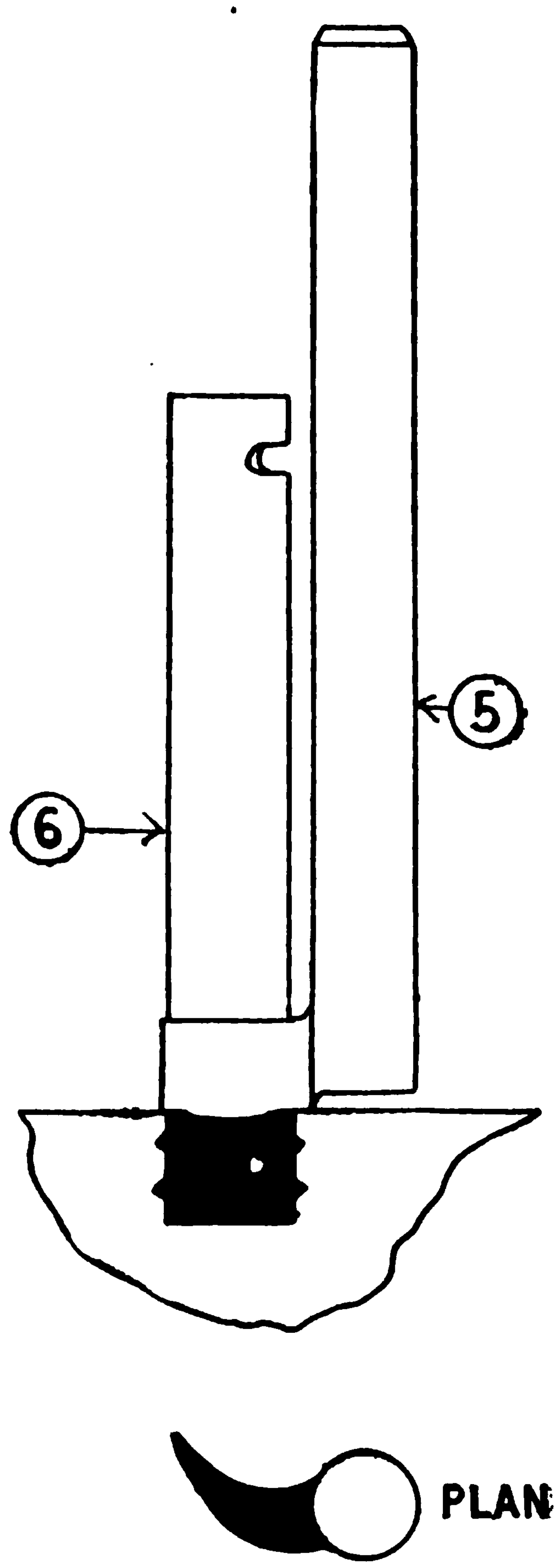
Allis-Chalmers or Willans-Robinson Type of Shrouded Blading.

NOTE.—The above system is not much in use for marine practice.

performed by vertical caulking tools and hammers of varying weight, according to the size of packing pieces which are being caulked. A portion should be caulked, and then a space missed, then another portion caulked, coming back again on that portion missed. If the caulking is carried right round it will tend to make the last segment rise out of the groove. Usually the blades are given three blows with hammer, heavier hammers being used on larger blades.

After the rotor is completely bladed it is now put into the lathe for tipping points of blades. The rotor is revolved at maxi-

mum speed of lathe, and the blade rows trued up by means of pointers fixed in line with the blade row being tipped ; flat pieces of wood which go in between rows of blades are used, and blades are pressed one way and another until row is running true. It is best to true up a portion of row square from the rotor drum. This

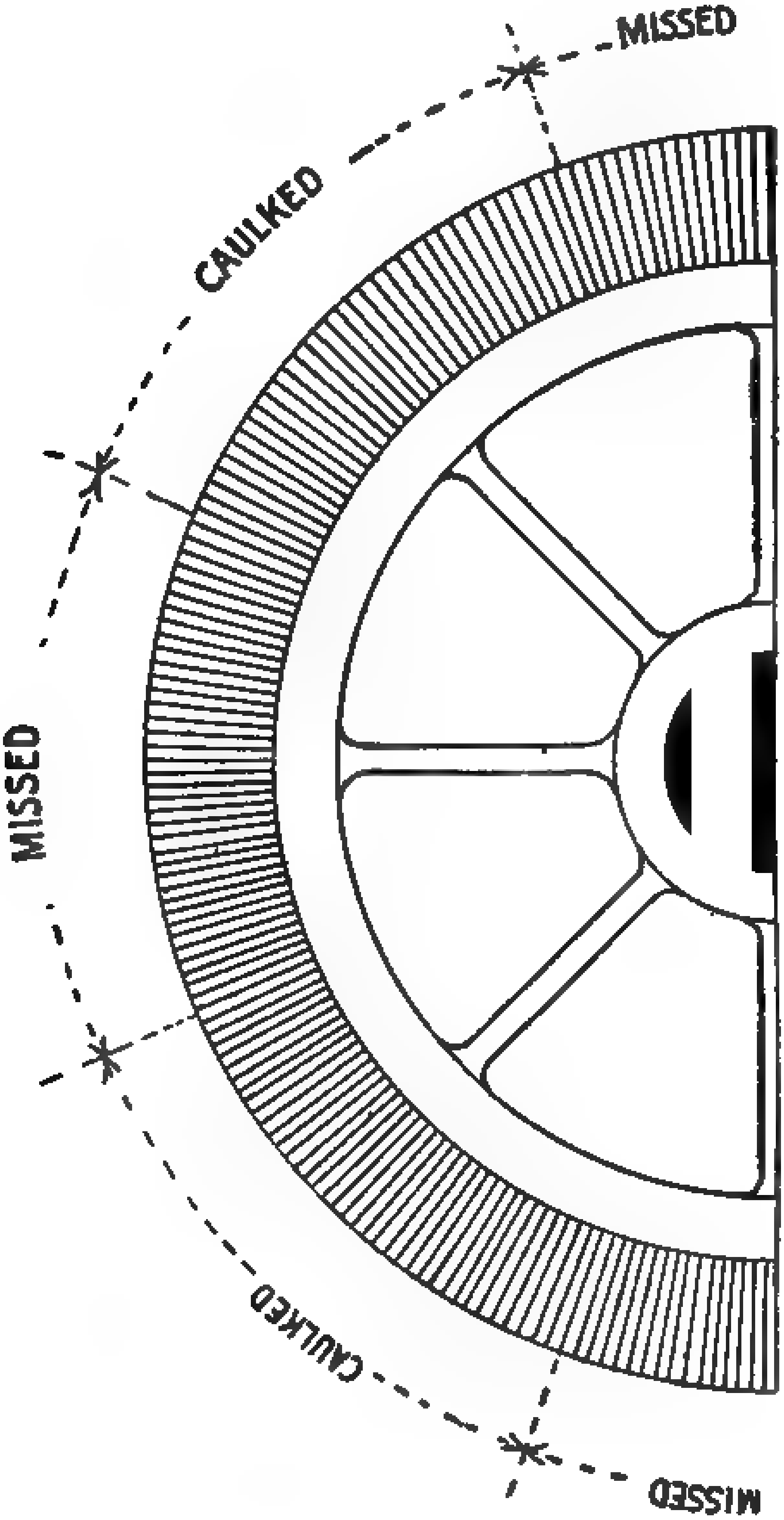


Segment Blading.

5. Caulking Tool shaped to clear Binding Wire. 6. Blade.

serves as a guide in trueing the rest of the row. The tipping is done by means of a tool having a sharp razor edge, and a very light cut is taken over tips of blades until they are reduced to the required diameter. This operation is important, as on it depends the radial clearance of the blades.

Casings.—The casings are first prepared for blading by the fitting of stopping pieces. These are made of soft brass, and their purpose

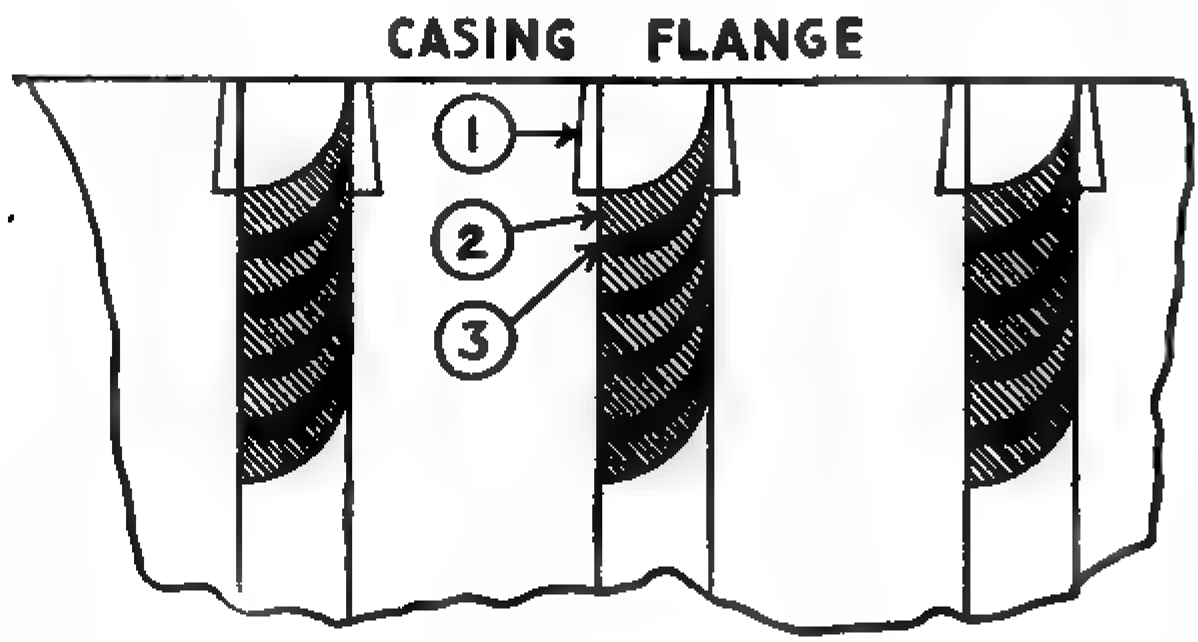


Method of Blading by "Segment" System.

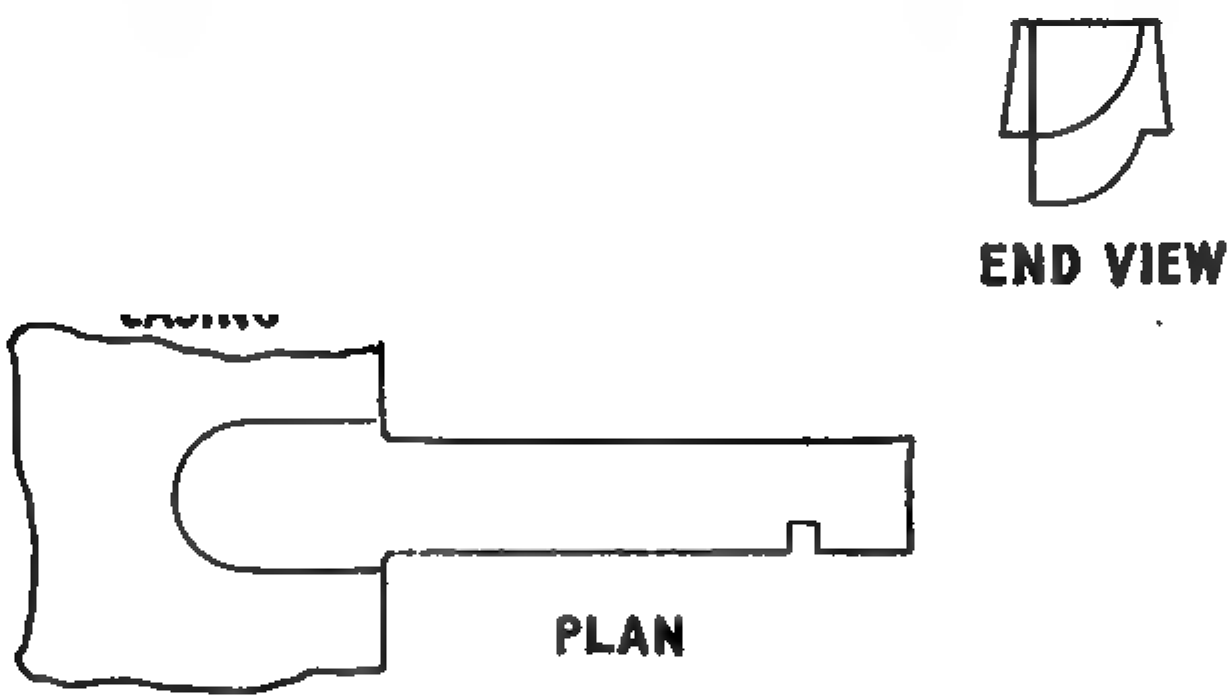
To ensure correctness of spacing and equal number of blades per row, the final caulking-in is completed as shown.

is to act as a stop for the blades at the joint of the casing. The prooves at the joint are dovetailed, and into this dovetail the stop gieces are fitted. This dovetail varies in depth and width according

to the size of blades of which the expansion is composed. After dovetails are cut, which is usually done in a milling machine, the stoppers are fitted into same, care being taken that they bear properly



View of Lower Half Casing (looking inside).
1. Blade Stopper. 2. Packing Piece. 3. Blade.

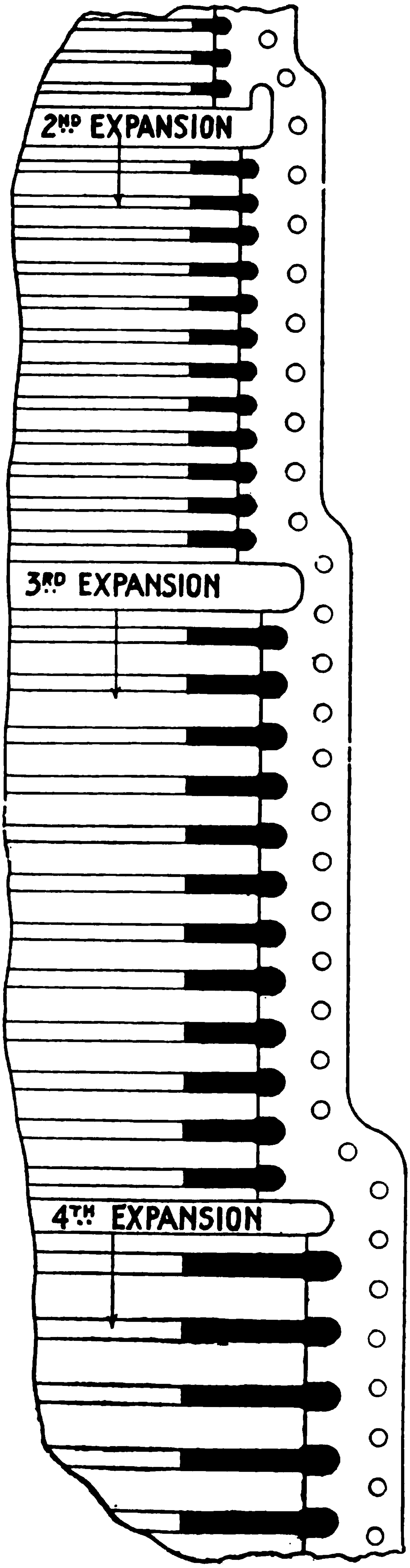


Casing Blade Stopper.

NOTE.—The stopper is fitted into the casing flanges top and bottom, and sets the angle of the blades.

Method of Blading Casings.

at the heel. One of the stoppers has a convex face of same shape as face of blade, and the other is concave, the shape of the back of the blade. The first segment is butted hard up against the stopping piece; the other segments to complete the row are butted against



Part of Turbine Casing showing Blade Stoppers in position.

this one, then the last segment is fitted in and stopper piece driven into place. At the end of the stopper piece a groove is cut in line with the binding wire on segment. The wire, which is left long for this purpose, is put into groove in stop piece and soldered in place. After all segments are fitted, the casing is now gauged across joint and stays put across casing; this is to prevent distortion of casing when caulking segments. It is usual to draw casing in to about .01 to .02 less than the finished diameter as it tends to keep the casing from distorting in the middle, to the same extent as it would if unstayed. The blades are caulked by caulking tools made so as to stand out clear of blades (see sketches). The point of the tool which strikes on packing piece is slightly rounded so as to deal the heaviest blow in the middle of packing piece. After this operation the stays binding the casing are taken off, and the casing gauged to ascertain the amount of distortion through caulking. The blades are now set up to length stick, which is set from the dummy face, and by means of wood battens the blades are squared up to same.

NOTE.—No permanent stoppers are fitted in the rotor grooves.

For good turbine efficiency results it is imperative that the angles, pitch, and position of the blades be as nearly mathematically correct as possible, any tendency to unequal circumferential pitch or angle (the blading being then known as "staggered") resulting in loss of efficiency.

The operation of "blading" being carried out entirely by hand labour, forms the most costly item in turbine construction.

The top of stopping pieces are milled flush to joint of casing, and the sides are also milled to exactly the breadth of the blades so as to ensure the same clearance all round the row. Both halves of casing having been dealt with in this manner, the horizontal joint is re-bedded, and casing is now ready for tipping of blades. The tipping is done in a horizontal boring machine by means of a tool similar to that before described in the tipping of rotors. After tipping, the blades are ragged by hand, *i.e.*, a small scraper is drawn across tips of blades so as to remove any rag or burr which may be formed on blades. It is advisable to lace the ends of segments together with copper wire so as to strengthen blades while undergoing the operation of tipping.

Difference in Number of Blades per Row.—In the blading of large rotors it sometimes happens that the number of blades per row varies by 5 or 6, owing to some of the "bladers" caulking the packing pieces further down than others, and so reducing the number of blades by increasing the space between each. As the process of blading is done entirely by hand the foregoing result is only to be expected, absolute uniformity of blading being almost impossible of attainment.

Stresses on Blades, &c.—It should be observed that the casing blades are subjected chiefly to a bending stress, due to the action of

the steam in striking the blade surfaces, whereas the rotor blades, in addition to a bending stress, are also subjected to a severe tensile stress set up by the centrifugal force of the rotor when revolving, and tending to throw out bodily the blades and packing pieces. To resist this, the grooves cut in the rotor drum and casing are serrated, as shown on page 87, to lock the packing pieces and blades in place.

The little stamped-out grooves shown near the bottom of the blades (both casing and rotor) allow of the packing piece being caulked in to lock the blades firmly in position.

Each "stopper" is of the same length as the corresponding blades. Regarding the materials employed for the blades, and clearances allowed, Mr E. M. Speakman says:—

"The material of which blades are usually made is a mixture of cheap brass containing about 16 per cent. of copper and 3 per cent. of tin. Alloys containing zinc are extremely unreliable for high temperatures, but blades containing about 98 per cent. of copper have been found very satisfactory for use with high superheats. More recently a material containing about 80 per cent. of copper and 20 per cent. of nickel has been adopted, and this is undoubtedly the best blading material existing. Steel blading, drawn in the same way as the usual brass section, has been used in the United States with fairly good results. The process of drawing turbine blades gives an extremely tough skin to the metal used, not only increasing the tensile strength, but greatly decreasing the chances of erosion.

"It seems probable that the usual caulking piece now adopted will be discarded in favour of a machine-divided strip into which the blades may be fitted, and instead of the slotting, wiring, lacing, and soldering process at the tip, a similarly machine-divided shroud will be used, giving a far stronger construction, and enabling finer clearances and better workmanship to be obtained; at the same time considerably reducing the cost of manufacture, and the risk of blade stripping.

"The chief causes of the latter may be set down to bad workmanship in fixing the blades, defective blade material, excessive cylinder distortion (this is probably the most fruitful cause, and is a serious one, being due to bad design), whipping of turbine spindles (which is also due to bad design, or bad balancing), wear of bearings (which is very remote), and the introduction of extraneous substances such as water or grit. In fact, blade stripping may be said to generally occur from preventible causes. Small vibrations of very high frequency occasionally set up an action in certain rows of responsive length that fatigues the blade material and causes the loss of blades without any fouling at all.

"Due to the action of the steam, an end thrust occurs in the direction of the propeller, which is advantageously used in partially balancing the propeller thrust, thereby reducing the size of the thrust block necessary. A margin must be allowed here, as the propeller thrust is not entirely balanced by the pressure on the annulus between the dummy-ring diameter D and the spindle, Fig. 5, C, plus the end pressure on the blades. For the diameter D to give the required annulus, as well as that of the propeller, the effective thrust must be carefully calculated; and experience shows that there is a drop in steam pressure varying from 10 to 15 lbs. per square

inch between the pipe inlet to the H.P. receiver and the first row of blades, which should be considered in designing this balancing area. The number of rows of dummy packing used varies according to the designer's judgment very largely, and may be modified according to the pressure and the clearance allowed—say a 7-1000th to a 15-1000th of an inch in electrical work, and rather more in marine work.

"The dimensions of the astern turbine are arrived at in the same manner as those of the ahead, the efficiency being largely sacrificed on account of weight and space; generally, the mean diameter is made practically the same as that of the H.P. drum.

"To a large extent, the inferior manœuvring capabilities of the earlier turbine steamers were due to insufficient astern power.

"It may be remembered that in a marine turbine the spindle is in compression and the cylinder in tension when working. In electrical turbines where the end thrust must be eliminated by the use of balancing pistons, the spindle is in tension and the cylinder is balanced. The shafts between the turbine bearings and the drum must be made amply stiff enough, as well as strong enough, for any sag in the spindle will destroy the clearance. The stresses due to centrifugal force are very low in the Parsons turbine, and except in occasional L.P. barrels do not exceed about 7,500 lbs. per square inch, while at the H.P. end they are usually under 2,000.

"The pressure on the bearings in a turbine is only due to the weight of the spindle, plus the negligible addition in marine work of that due to any gyroscopic action; it may be taken as from 80 to 90 lbs. per square inch as long as the rubbing velocity does not exceed 30 feet per second. If it does, the pressure must be reduced so that the product of pressure × velocity does not exceed 2,500-2,700. In land work, 50 lbs. × 50 feet is very common. The friction heat of the bearings added to that due to conduction through the pedestals necessitates the use of large oil coolers, and in the case of very high temperatures, of special kinds of oil. If possible, the bearing temperature should not exceed from 140° to 150° F., though the writer has known of 190° F. being used without trouble. In marine turbines this temperature is usually much lower. Rigid bearings are used for marine spindles, not the flexible type adopted in land work.

"Space does not permit of more than passing reference to cylinders; but it would be difficult to exaggerate the importance of very careful design in this connection. Cylinders, with heavy flanges on the centre line, distort in a very curious fashion when heated with their axis horizontal, and measurements taken off a hot cylinder on a surface plate with micrometer gauges reveal some very remarkable facts. When working, the temperature along the cylinder falls possibly from 400° to 100° F. in a distance of 6 or 8 feet, and, unlike the reciprocating engine, this remains constant; the radial expansion is consequently more at one end than the other; while at any point along the turbine the tendency is to expand less at the flanges than at the top and bottom. For this reason ample clearance must be allowed; exactly what this will be when spindle and cylinder are hot is hard to say, but it seems most likely that the total clearance area will differ but little from what it is when cold.

"The longitudinal expansion when hot is often very marked, and in all turbines necessitates provision for the resultant movement at one end. In marine work the after end of the cylinder is secured to the vessel, the engine

seating also performing the function of a thrust-block seat, while the forward end slides forward, taking with it the entire shafting. The thrust block is at the forward end of the cylinder, and also performs the duties of an adjustment block for setting the longitudinal clearances, to do which generally necessitates uncoupling the shafting abaft the turbine.

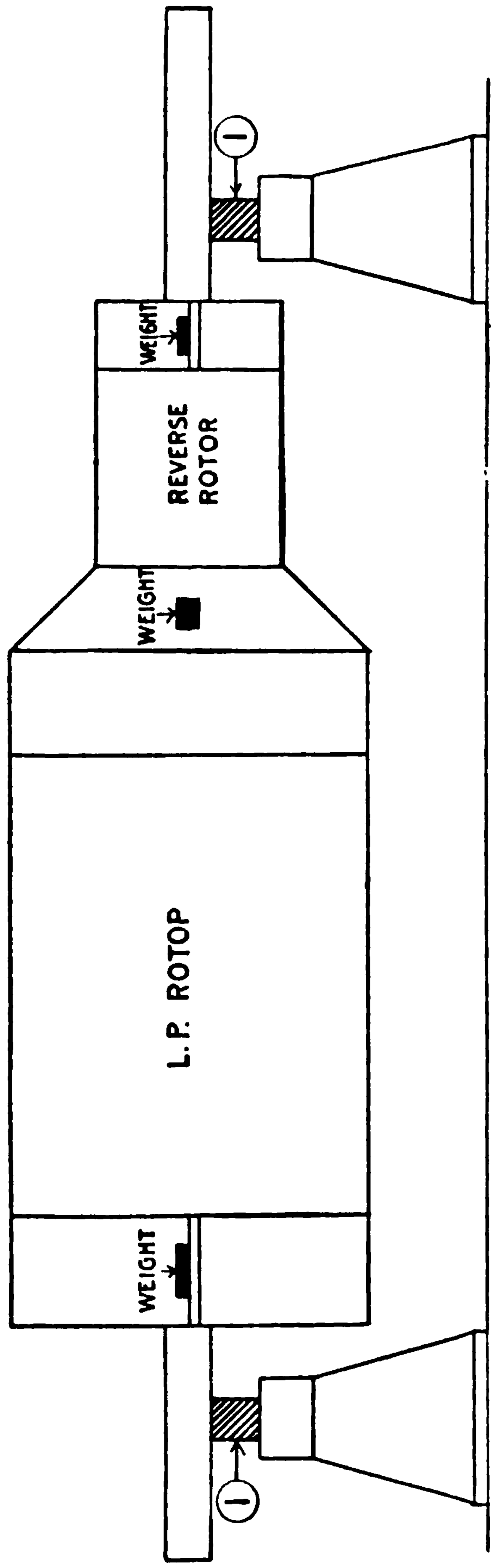
"The difference in expansion between the cylinder and spindle, from the thrust block to the dummy ring, may be the cause of serious difficulties in large marine turbines, unless the closest attention is paid to this feature in the design; and 'warming up' with these large cylinders needs possibly even more care than is essential with large piston engines.

"On shipboard, the turbine cylinders are practically under one's feet, and the radiation from them is very unpleasant, especially if there is any leakage from the glands. To all who are responsible for the lagging of cylinders and the system of ventilation in turbine engine-rooms I would call attention to the possibility of their having to stand a watch of four or six hours on the top of the H.P. cylinder, such as is the case in the 'Eden' or 'Amethyst,' the heat in the latter vessel being almost unbearable. With reciprocating engines, one stands on a comparatively cool lower platform with the cylinders overhead, and with some chance of the hot gases rising clear, but in naval turbine work under a low deck this point has not met with adequate attention.

"In the course of operation, more especially in marine work where no superheaters are used, there is a distinct tendency for the turbine to be supplied with wet steam, the effect of which on the economy is very marked. Experiments that have been made show that the percentage increase in consumption is about twice that of the moisture in the steam. For instance, with 2 per cent. of moisture in the steam at the first row, the consumption is increased about 4 per cent."

Balancing of Rotors.—Two balances are required to ensure a steady running rotor: these are the static balance and the dynamic balance. The static balance is obtained by revolving the rotor by hand on a pair of knife edges or balancing stools, which consist of two supports of cast iron, having on top a truly machined face of hard grained cast iron. This face should be 2 to 3 in. in breadth, according to weight of rotors which are likely to be balanced. The stools are levelled up longitudinal, and to each other, after which the rotor is laid on these, resting on the bearing part of the shaft. The rotor is revolved and allowed to come to rest, and the point at the bottom will be the heavy part. Temporary plates are fixed on to the wheel opposite this, and the rotor revolved and weights added until it will stand at rest in any position. This being arrived at, the next operation is to determine the distribution of weights, which is a most important part, as on this may depend the success of the static balance in relation to the dynamic balance. If it is an H.P. rotor, having two wheels of the same diameter—one forward and one aft—it is usually best to put one-half of weight on each wheel: if, however, an L.P. and astern rotor combined is being balanced, having three wheels—two of the same diameter, and the after one smaller—it is advisable to put a slightly heavier weight on the after wheel than on the other

two, as owing to the smaller diameter of the aft wheel the weight will be acting at a smaller leverage. An important point is to put all weights in line if possible, and to have them duplicate as regards bolt



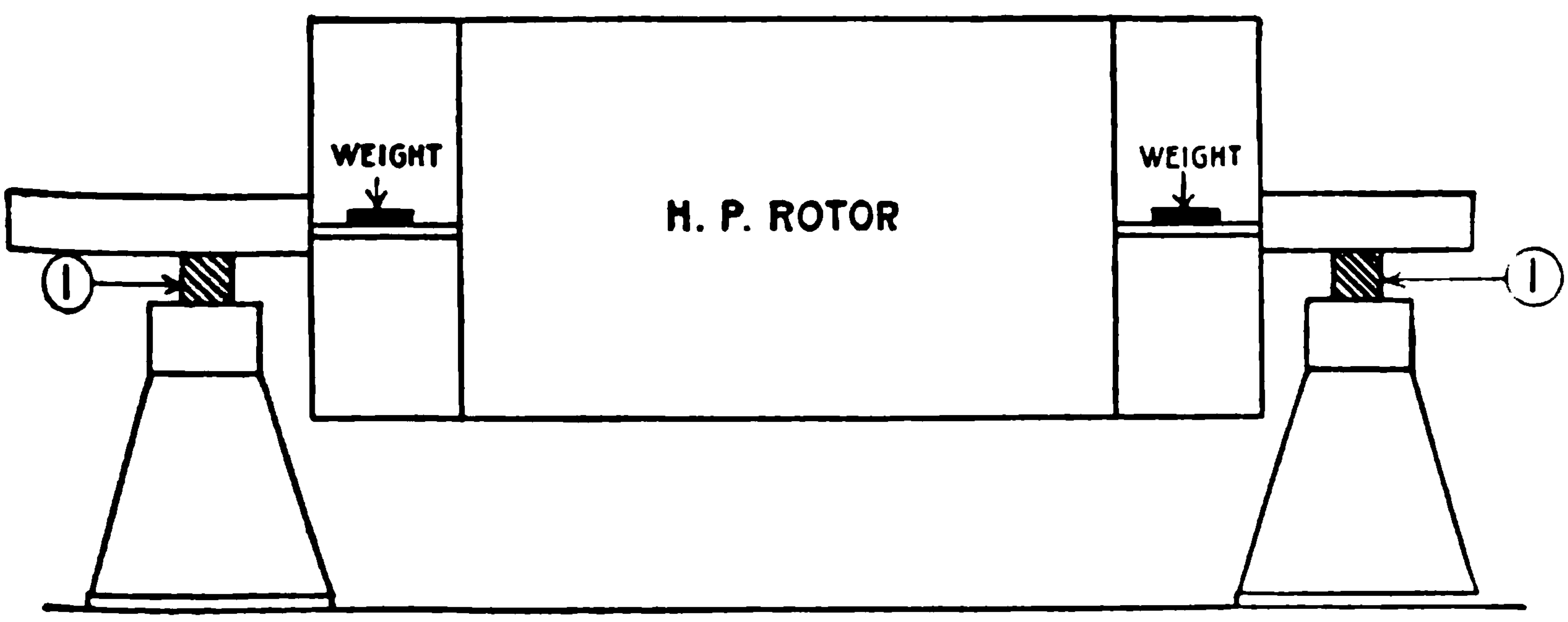
Static Balance of L.P. and Reverse Rotors.

1. Knife Edge.

holes. After weights are fitted the rotor is now tested for balance. This is done by dividing the circumference of the wheel into eight equal parts. A stationary pointer is attached to an upright batten fixed to a small knee plate which rests upon the floor, a line is drawn

across the rim of the wheel in line with the centre of the shaft, a small weight of, say, 1 lb. or so is suspended by a hook attached to the wheel; the rotor will move a small amount and then come to rest, when another line is drawn across rim of wheel: this distance is measured and note taken of same. The same operation is performed at the other seven points, and the result compared: if the rotor is truly balanced the movement should be the same at each point. If movement varies then weights will require to be altered to suit.

"Dynamic" Balancing of Rotors.—The turbine is balanced dynamically by one of two methods—(1) By being revolved by steam acting as under working conditions, and tested for vibration by fixed indicating pointers or pencils. (2) By some suitable arrangement of spring bearings in which the rotor is placed and rotated by some external power, such as an electric motor coupled up to the spindle, and as before a pointer indicates the vibrations on a piece of diagram paper by a more or less "waving" line. The latter method is the



Static Balance of H.P. Rotor.
1. Knife Edge.

most accurate, as the rotating force acts from the centre outwards, and, if anything, tends to magnify any lack of balance or difference in weight which the rotor may possess. At the same time, few rotors are actually in a state of perfect balance, as no really scientific and accurate method of dynamic balancing has yet been devised for turbines. Thin plates are either pinned on as required to those portions of the rotor which are found to be lighter and therefore require added weight, or metal is chipped off the "balance" strips on the heavier positions.

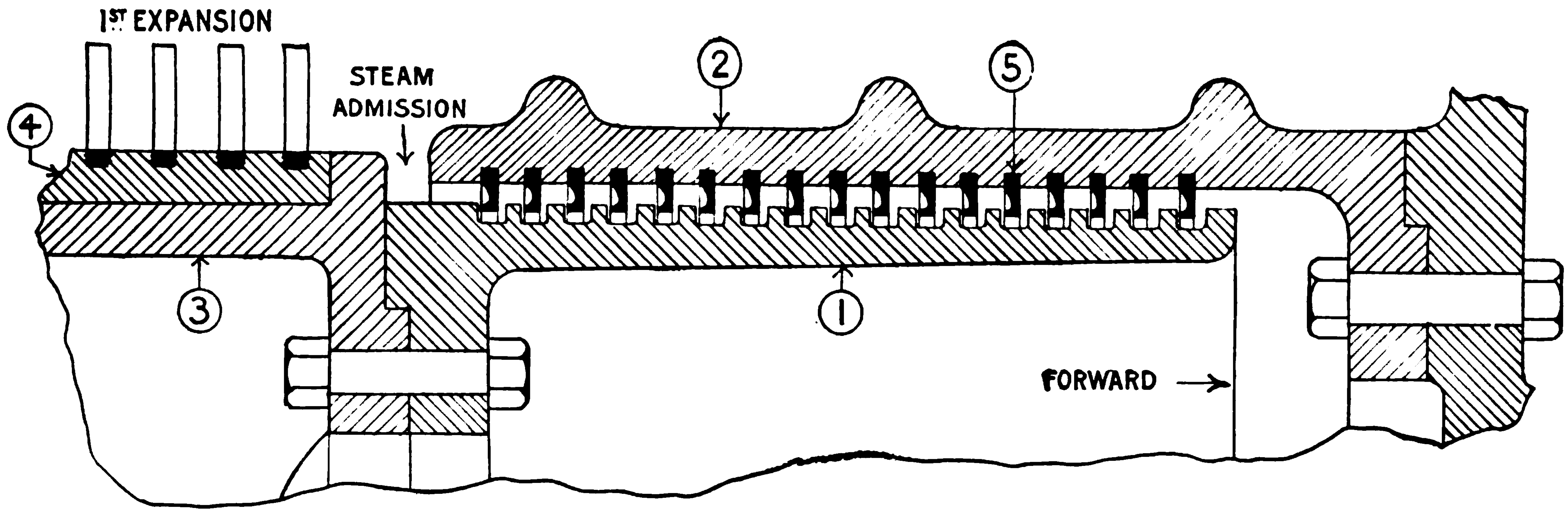
The rotor should be revolved to at least 20 per cent. above the required revolutions. Should the balance be imperfect it will show by vibrations being set up, and if the balance is very bad may result in damaging the blading of the turbine. When the rotor is at its worst point of vibration, marks should be put on shaft at forward and after end: it is important that these marks should be put on at the same time. The turbine is now stopped and the relation of the marks to each other examined. If the marks on each end are opposite, or not in line with each other, and one end in line with the weights and the other

opposite, then by taking a weight from one end and putting it on the other this will ensure the static balance being maintained and the dynamic balance improved, if not perfected. After balancing, the bolts through the weights are riveted over on the points to prevent slackening back. In running turbines under steam, especially when developing high speed, it is found that there is usually a point when vibration will begin to show—this is termed the critical point of speed. This is believed to be due to a change in the axis of rotation caused through deflection, or bending of the shaft, which causes the rotor to revolve round a new axis, termed the axis of deflection, and which alters the centre of gravity of the rotor and causes vibration. This may not occur at the highest speed, as the vibration sometimes decreases as speed is increased.

The various elements which go to make up the complete rotor such as drum, wheels, and spindles, are each carefully balanced before assembling, then the whole is again statically balanced on knife edges, after which an approximate dynamic balance test is employed by running the turbine under steam and noting the vibrations at either end.

The rolling balance is tested by noting if the rotor will stop dead at any position when moved by hand along the knife edges. If not, the balance may be first adjusted by sticking on pieces of clay at the lighter positions, and when the balance is thus corrected, the clay is replaced by metal plates which are pinned on to the wheels. With some firms the practice is to fit the wheels with eight balance strips which are cut off as required to lighten the heavy side of the rotor, in place of adding weights to the lighter side. The extra weight of the balance strips may amount to about 30 lbs. in fair sized rotors, and may be thus reduced to suit the balance as required. When testing the dynamic balance by running under steam, it is advisable to leave the static balance undisturbed, and this is arranged for by merely taking weight from one end and giving it to the other end, keeping of course the same circumferential position. When the dynamic balance is out, it is indicated by vibrations at the heavy end, and, as before mentioned, this is corrected by transferring weight from one wheel to the other. No system of calculation is employed in either balances, and the excellent results generally obtained are entirely the result of experience in the work together with good workmanship.

Dummies.—The dummies are placed at the steam admission end of each turbine, and two kinds of dummies known respectively as radial and facial are employed: the facial dummy is usually fitted in the ahead turbine, and the radial dummy in the astern turbine. The principle of the dummy is to prevent the steam from escaping through the interior of the rotor to the exhaust end of the casing instead of doing its legitimate work in passing through the blades of the turbine. Another reason is that if no dummies were fitted, the full initial pressure would be on the glands instead of exhaust or terminal pressure. A facial dummy consists of two parts called the casing dummy and the rotor

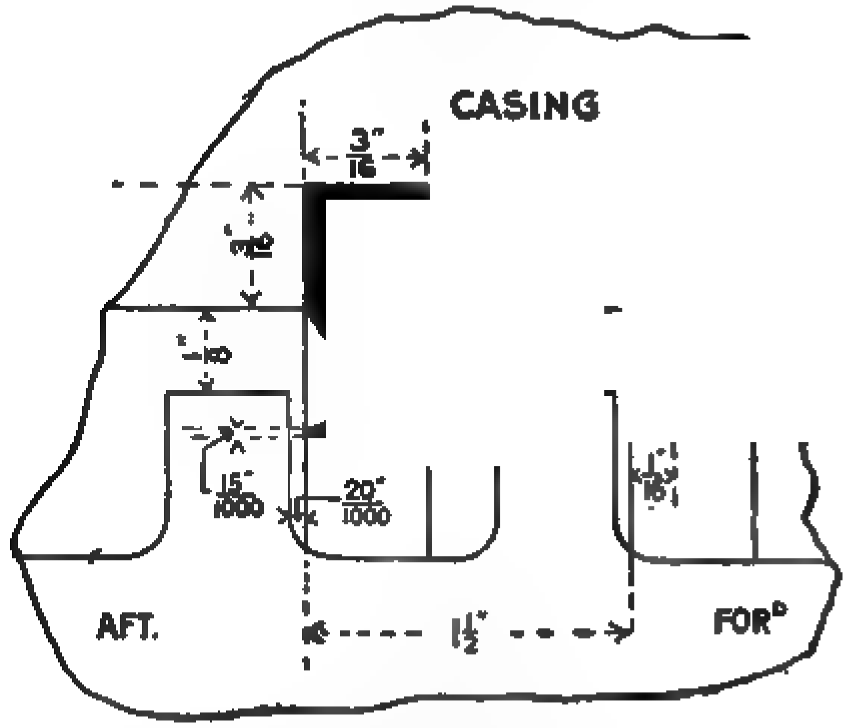


Ahead Type of Dummy (Facial).

1. Rotor Dummy. 3. Wheel Drum.
2. Casing Dummy. 4. Rotor Drum.
5. Undercut Brass Rings bedded into Casing.

The difference in diameter between the rotor dummy and drum, as shown, forms the annulus upon which the steam acts to partly counterbalance the propeller thrust.

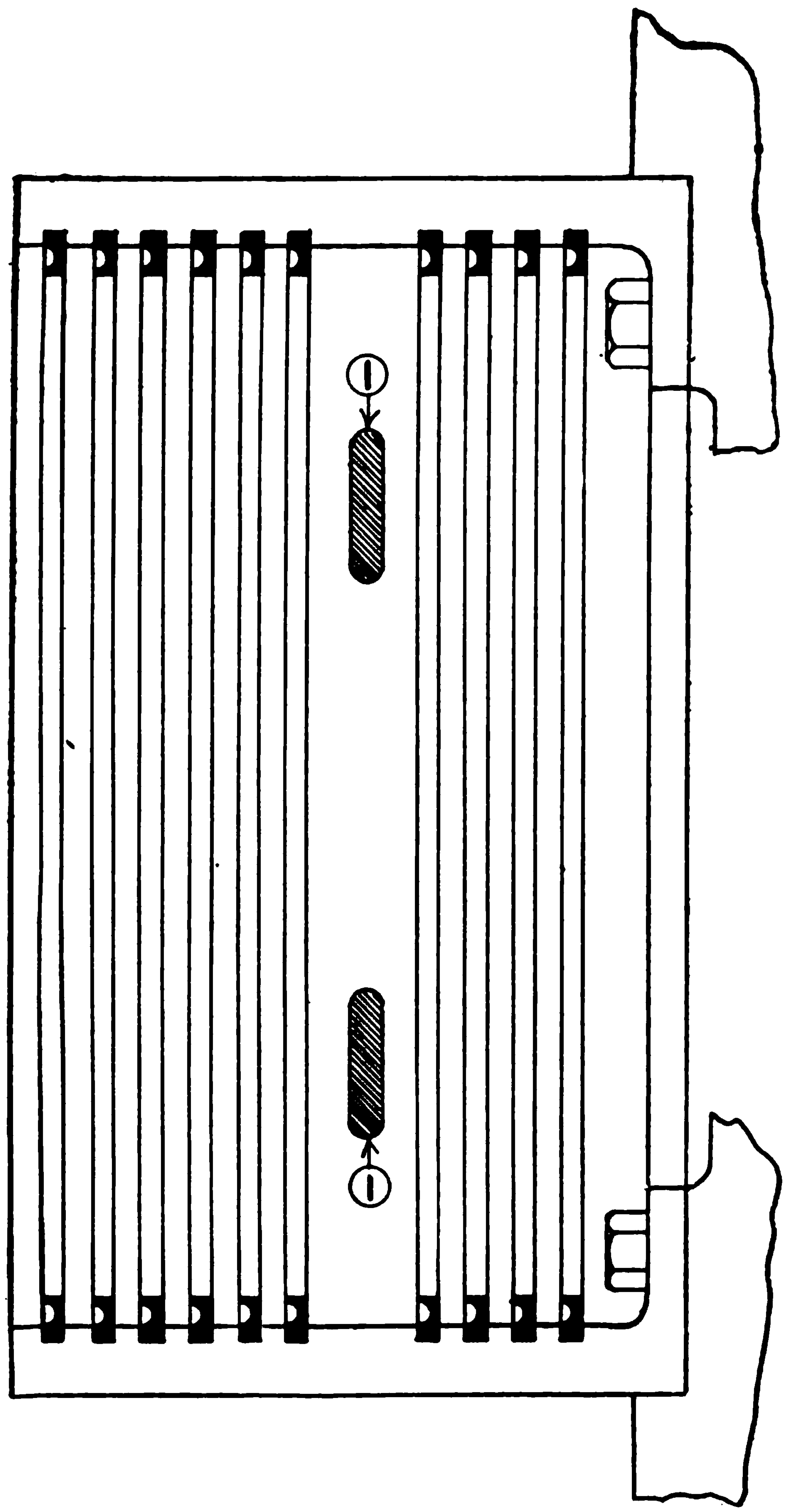
dummy. The casing dummy is a cast-iron cylinder, which is in two halves, bolted together at the horizontal joint. This cylinder is bored out and grooved, the grooves being usually $\frac{1}{8}$ in. wide and $\frac{3}{16}$ in. deep, and into these grooves are driven brass strips. The brass strips are bent to the radius of the cylinder, and a serration made in them, so that the serration is just flush when the strip is driven into groove. After the blades are in place, the metal of the cylinder is caulked into the serration, thus binding the strips. The strips are cut in lengths of 6 in., and at each end of the half of the casing in one row a 9-in. piece is put in, and in the succeeding row a $4\frac{1}{2}$-in. piece is put in, so that the joints in each row are not in line. The strips are left ".012" clear of each other at the ends, so as to allow for expansion. After the blades are all in place and caulked, the dummy is put into lathe, and the blades turned up.



Ahead Dummy Facial Rings.
(With average dimensions.

The blades have a face bearing of ".015" so as to ensure that if the rotor dummy should touch, the friction caused thereby would be reduced to a minimum. The rotor dummy is of steel, and usually is a round cylinder, although sometimes it is made in two halves. The dummy is rigidly bolted to the rotor, and turned up in place. A series of grooves corresponding to the brass strips in the casing dummy are turned out, having a fillet in both sides of groove, the grooves being $\frac{3}{16}$ in. deep. The brass strips in the casing dummy project into the groove in the rotor dummy $\frac{1}{8}$ in. When the rotors are set to position in the casing, the factor which determines this position is the dummy clearance, this varying according to the size of turbines. For average sizes the clearance is usually as follows:—

.015 to .020 in the high-pressure turbine,
and .020 to .03 in the low-pressure turbine.

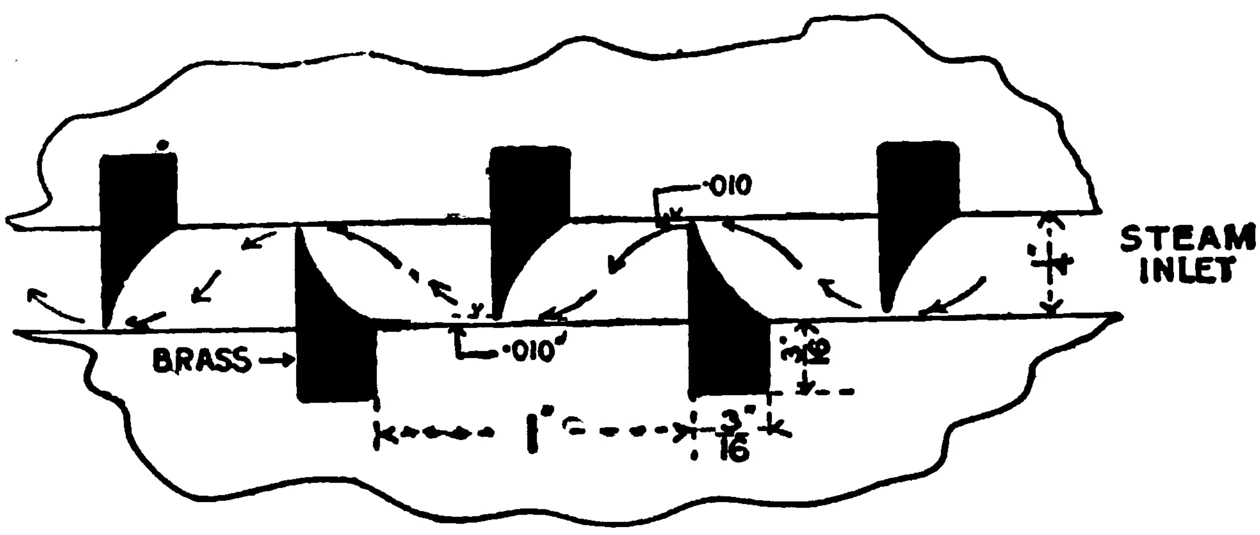


Ahead Dummy Casing.

1, 1.—"Leak-off" Ports.

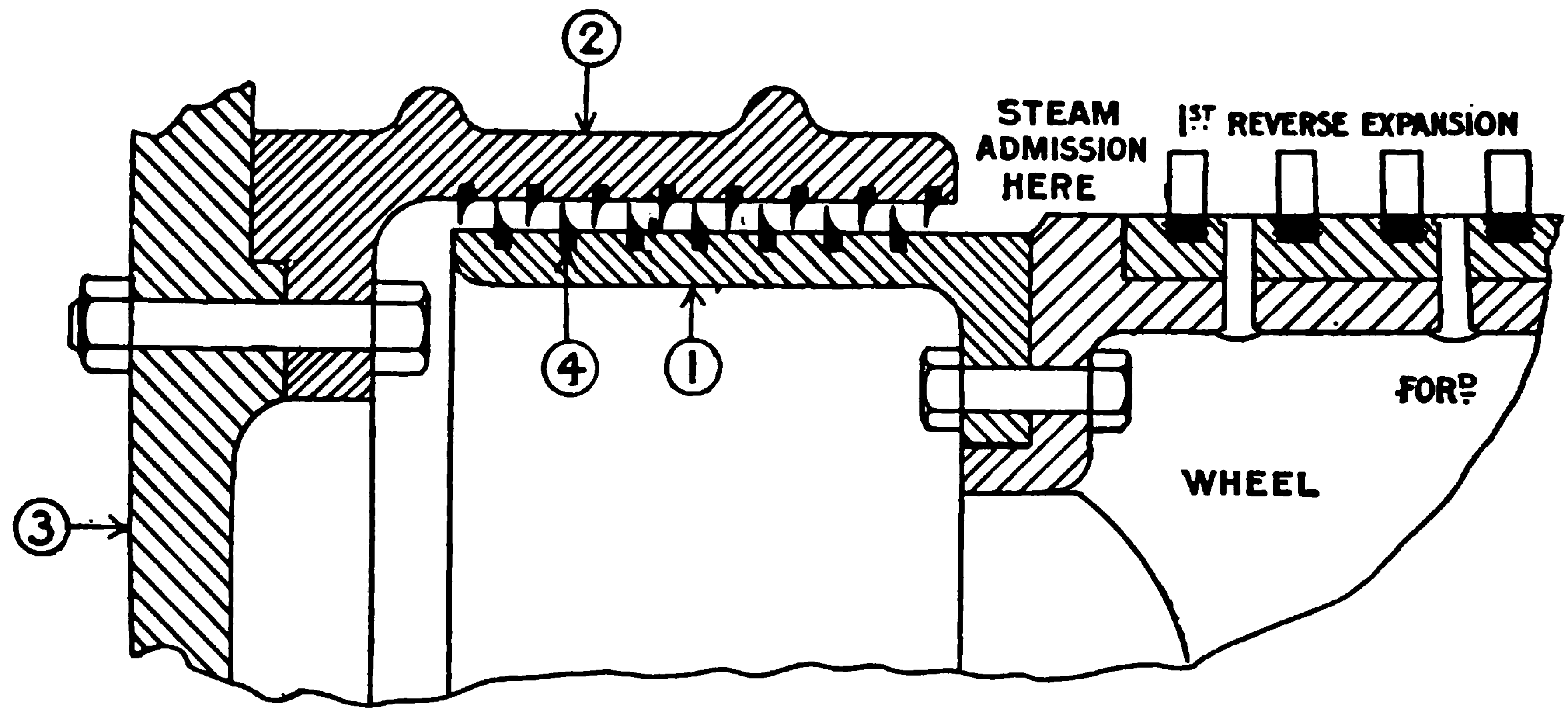
NOTE.—Usually not less than 20 of the small brass rings shown are fitted to each ahead dummy. The rings are formed of $\frac{3}{16}$-inch brass at a pitch of about $\frac{1}{2}$ inch. The "leak-off" ports shown are omitted in later designs.

Radial dummies are generally fitted in the astern turbine, and as in most turbines the astern rotor is part of the L.P. rotor, the expansion under steam takes place in an aft direction; it will thus be readily seen that a face dummy would not be possible. A radial dummy consists of two parts, the casing and rotor dummies. The casing dummy is a cast-iron cylinder bolted to the end of the turbine casing, and has grooves cut the same as a facial dummy; the grooves are



Radial Fins of Astern Dummy.
(With average sizes.)

bladed in the same manner with brass strips, and the strips are turned up to a knife edge, with one straight side, and the other side with a fillet as shown in sketch. The rotor dummy is also grooved, and in this case brass blades are also put in; these blades project



Astern Radial Dummy.

1. Rotor Dummy. 3. End of Casing.
2. Casing Dummy. 4. Radial Fin Rings.

NOTE.—The number of rings usually fitted varies from 6 to 8.

in between the blades on the casing dummy, but do not touch, and the clearance allowed from the tip of blades is usually .015 to .020. The steam escaping through the blades is first wire drawn, and then expanded, and so on until it escapes at the end into the interior of

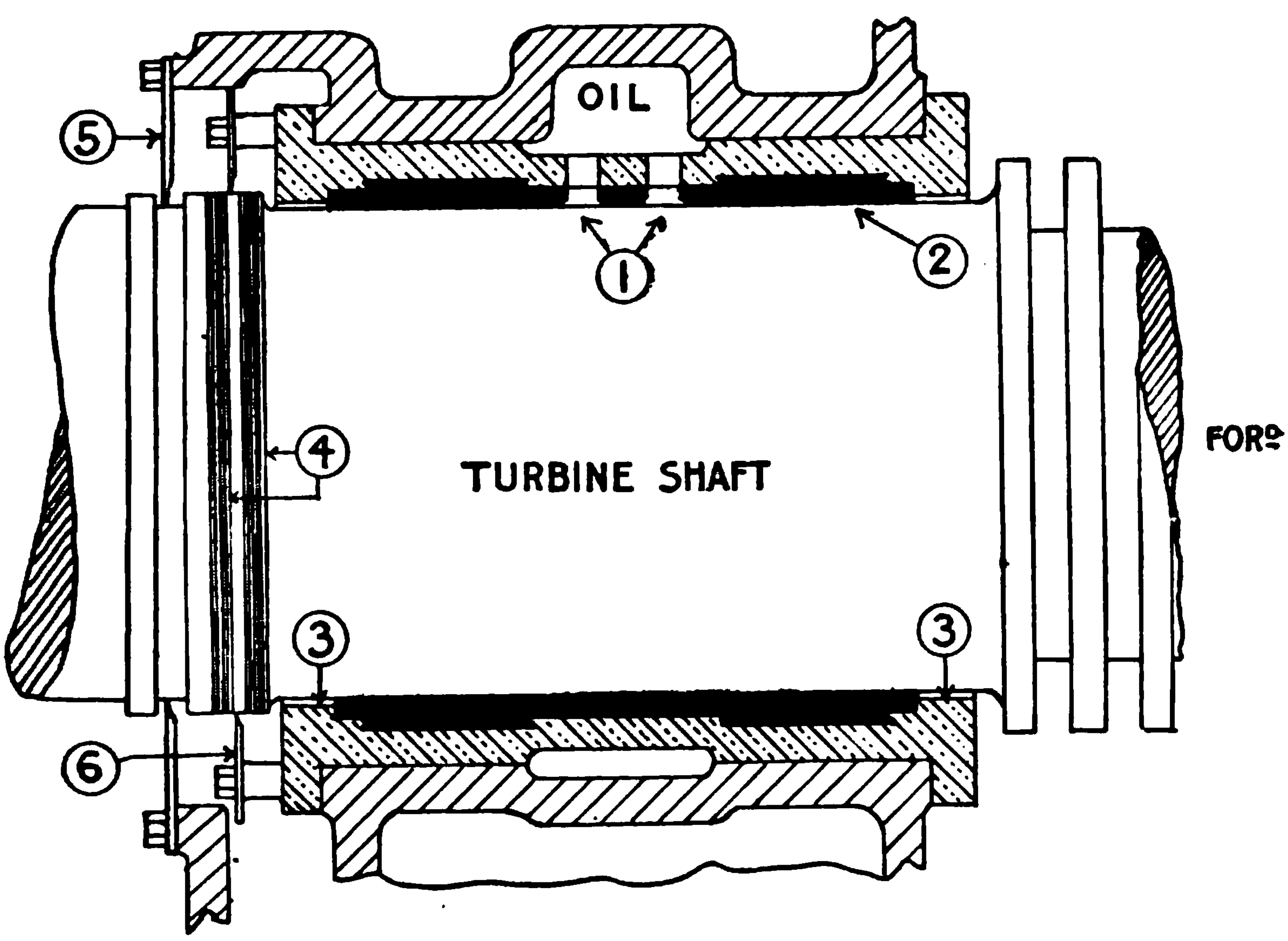
the rotor, which is in a vacuum if a low-pressure astern dummy and to the exhaust pipe leading to the L.P. if a high-pressure astern dummy.

Dummy "Leak-off."—The ahead dummies are sometimes supplied with a "leak-off" from the space between the sets of rings to the 3rd expansion of the same turbine.

Any steam which finds its way past the inward set of rings passes away by the "leak-off" ports and pipe to one of the other expansions where the pressure is less (see sketch).

As before stated, the "dummies" prevent steam leakage at the high-pressure ends of the rotor and act so as to produce (by means of wire drawing off the escaping steam) a "water seal" by the resulting condensation.

Main Bearings and Adjusting Blocks.

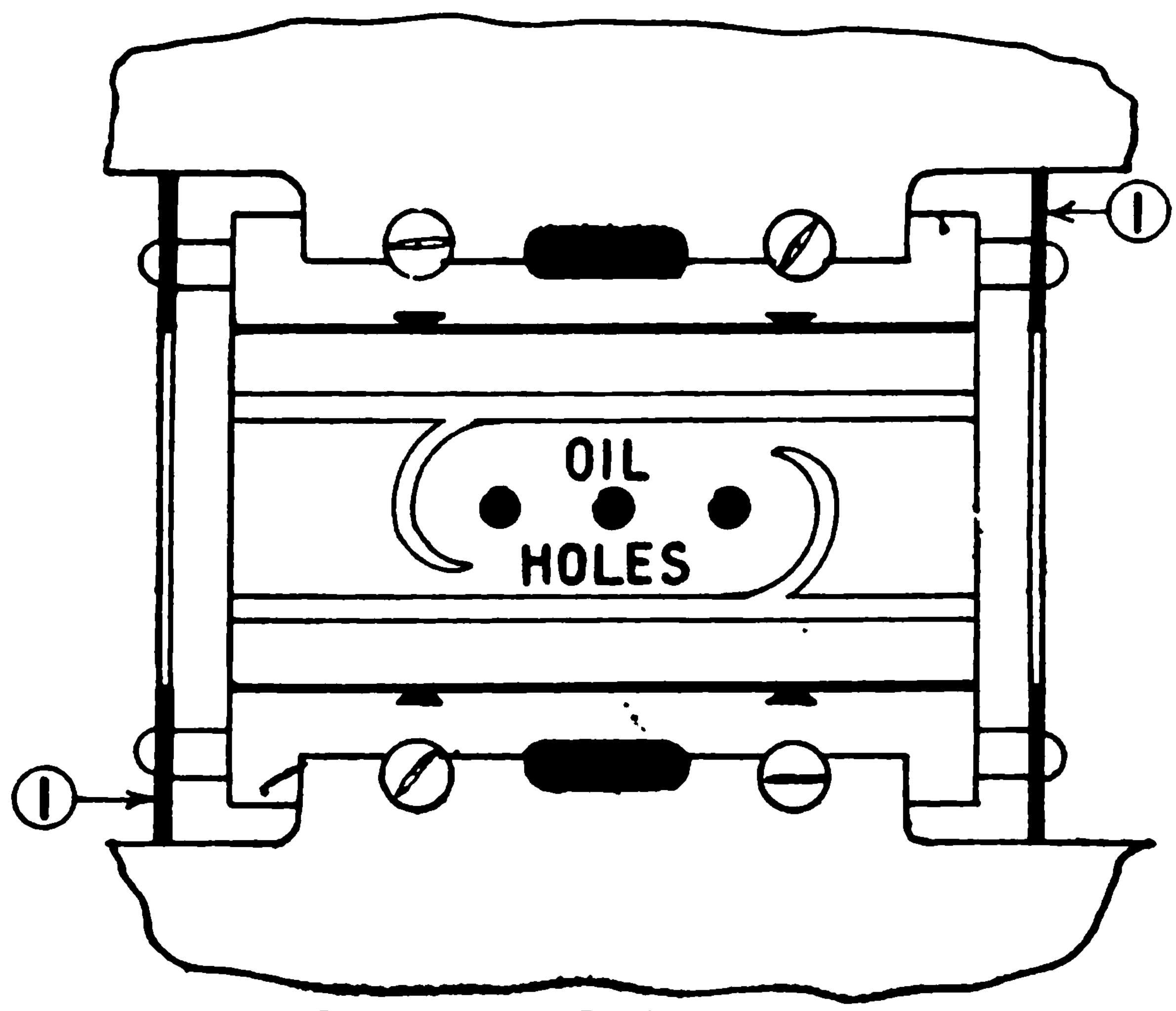


Plan of Main Bearing.

1. Oil Holes. 2. White Metal. 3. "Reliefs," $\frac{15}{1000}$ in. clear, to prevent damage to blades if white metal runs out. 4. Oil Baffle Grooves. 5. Oil Baffle Plates.

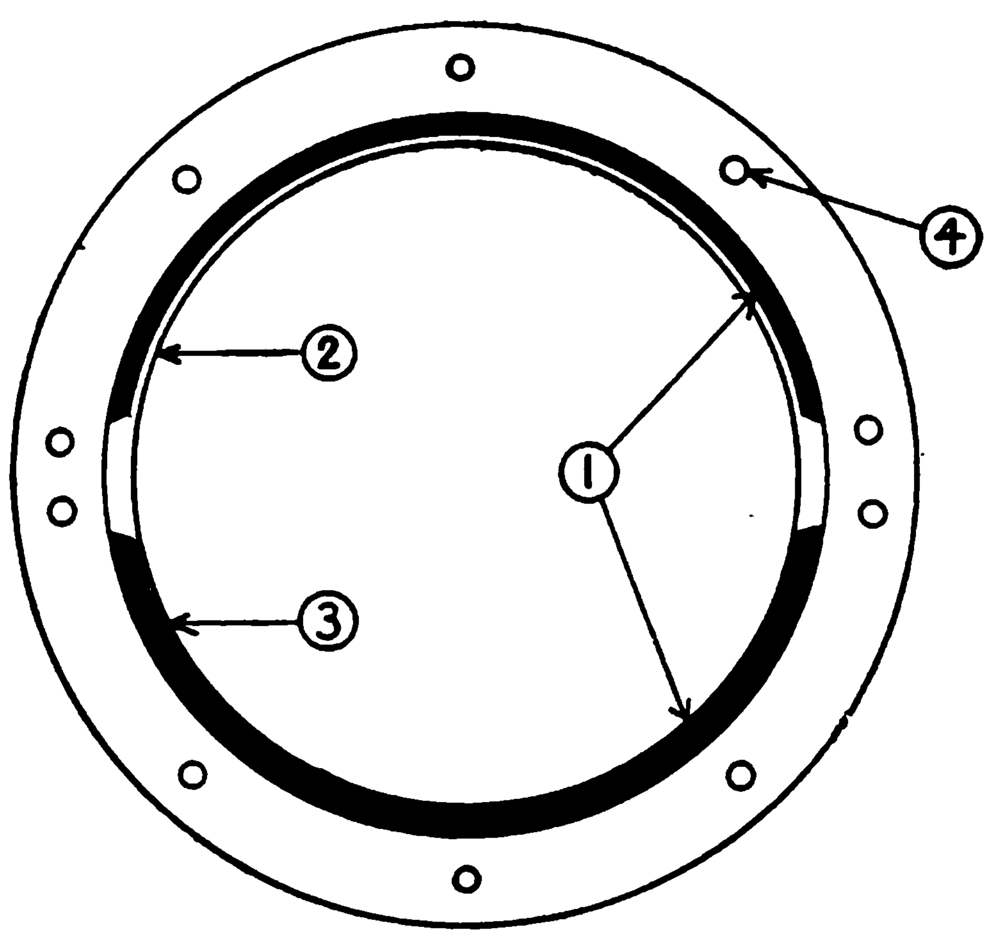
Main Bearings.—A bearing is fitted at each end of each rotor, and is of the usual marine type, being of either brass or cast iron (the former for naval work), filled up with white metal. The bearings are held in place by four large screws, each one being partly in the seat and partly in the brass as shown. Small holes are bored through the top or bottom half of the brass, and the oil is forced through the oil holes by the oil pumps, as elsewhere described. The pressure on the bearings is usually from 70 to 80 lbs. per sq. in.

Oil Baffles or Deflectors.—Ring plates of brass in two halves are secured to each end of the after brass, and to the after end of the



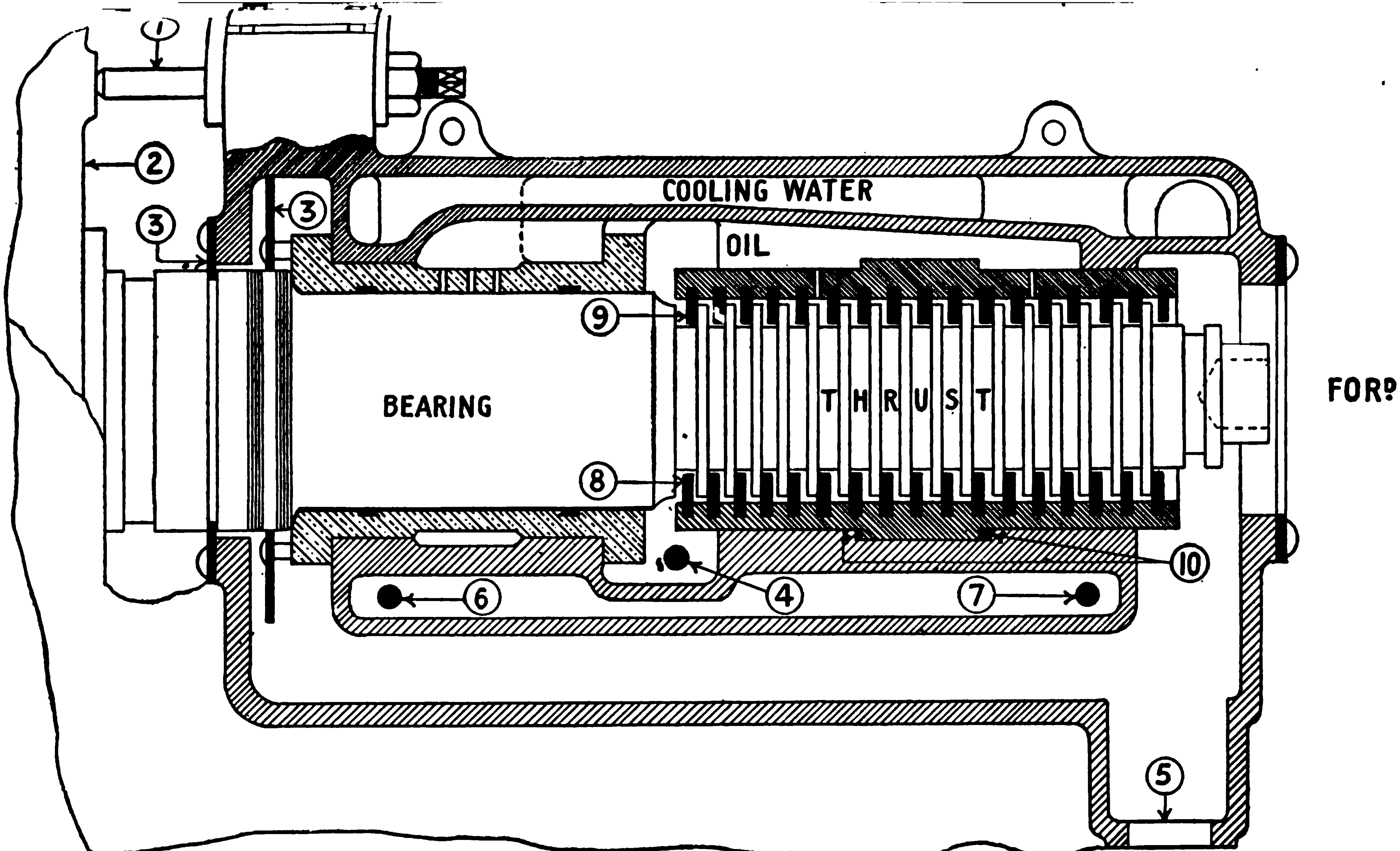
Lower Half Main Bearing.
1, 1. Oil Deflecting Rings.

forward brass, to act as oil baffles, that is, to prevent the ingress of oil to the rotor casing and blades. These baffle ring plates fit in



End View of Main Bearing Bush.
1. White Metal. 2. Clearance of .015 in. 3. Clearance of .004 in. 4. Baffle Plate Studs.

very closely round the spindle circumference and thus limit the oil to the required positions. These are usually of $\frac{1}{16}$-in. or $\frac{1}{8}$-in. brass plate, the plates being knife edged and fitted .004 clear of shaft. The

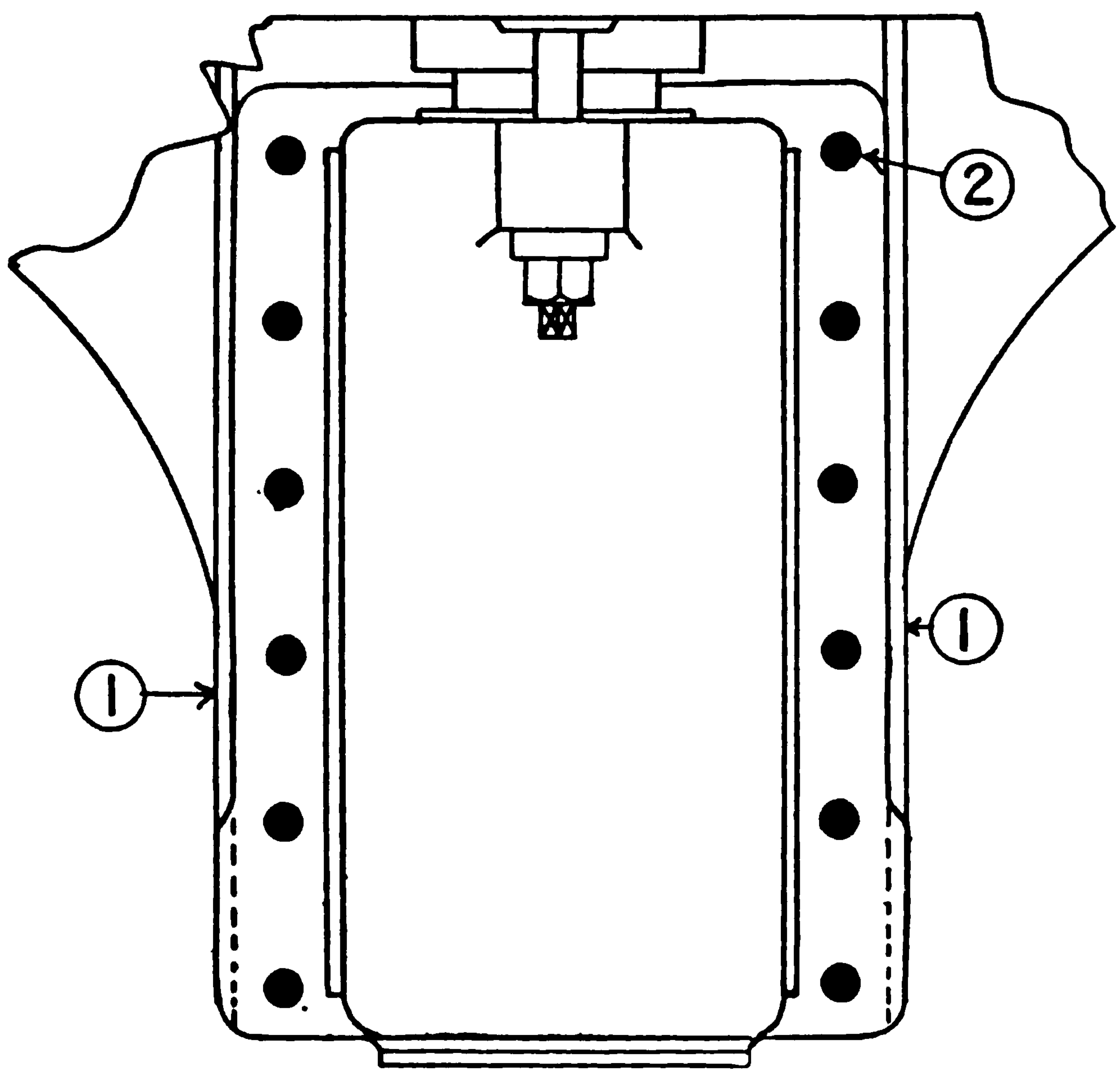


Sectiona! V.ew of Forward Bearing and Thrust.

Adjustment Stud. 2. End of Turbine Casing. 3. Oil Deflecting Rings. 4. Oil Inlet (from oil pumps). 5. Oil Drain to cooling tanks. 6. Cooling Water Inlet. 7. Cooling Water Outlet. 8. Ahead Thrust Half Rings. 9. Astern Thrust Half Rings. 10. Thrust Adjustment Half Rings.

top half of the bearing is also lined with white metal, and is usually le .015 clear of shaft to allow for passage of lubricating oil, and also to allo for expansion of shaft when heated up. It is also advisable to hav bearings .008 wider in diameter across the sides for the same purpos

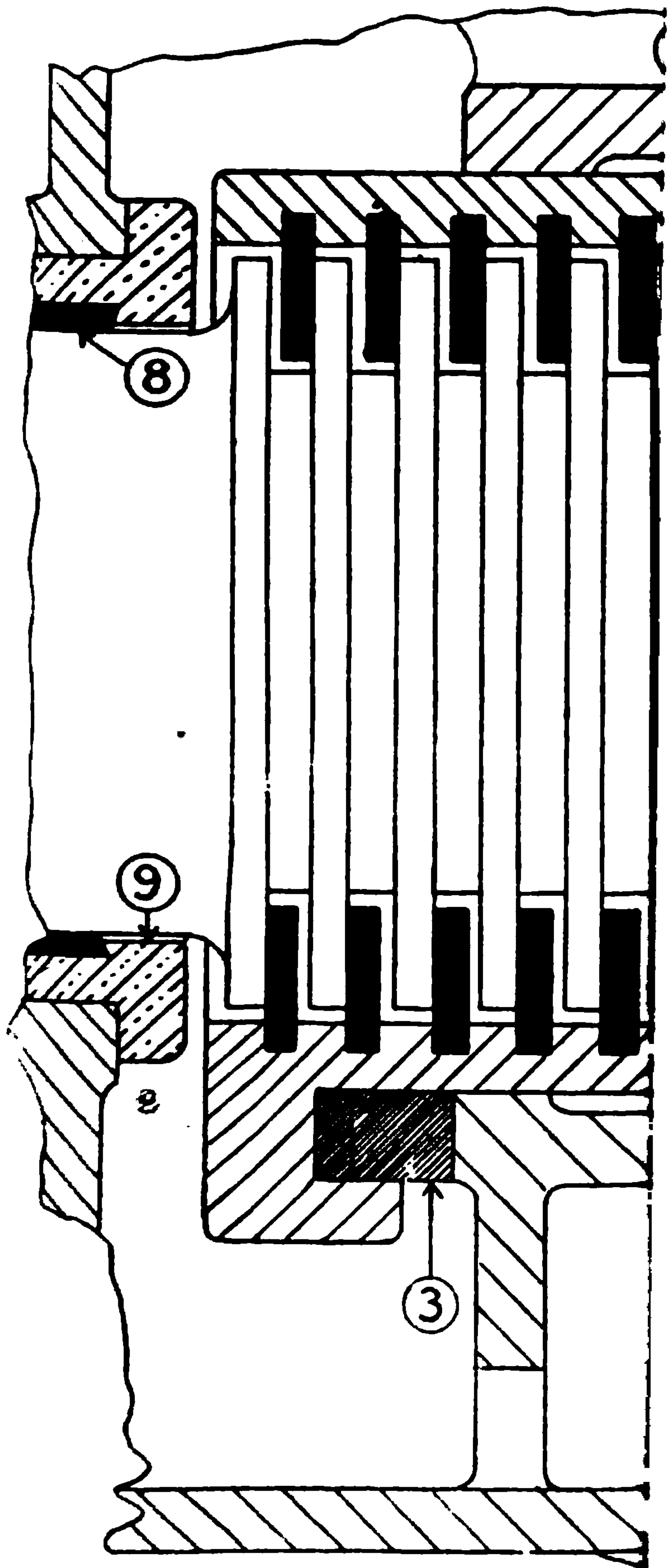
Thrust Block.—The adjusting block in the Parsons turbine consis of a forged steel brush grooved out for receiving brass rings. The thru usually consists of seventeen rings. The rings are fitted into plac and the steel is caulked into a serration on the rings. Through th rings, with the exception of the two end ones, holes are bored, usuall three in each ring, and gutters are cut from these to the edge.



Plan of Thrust Cover.

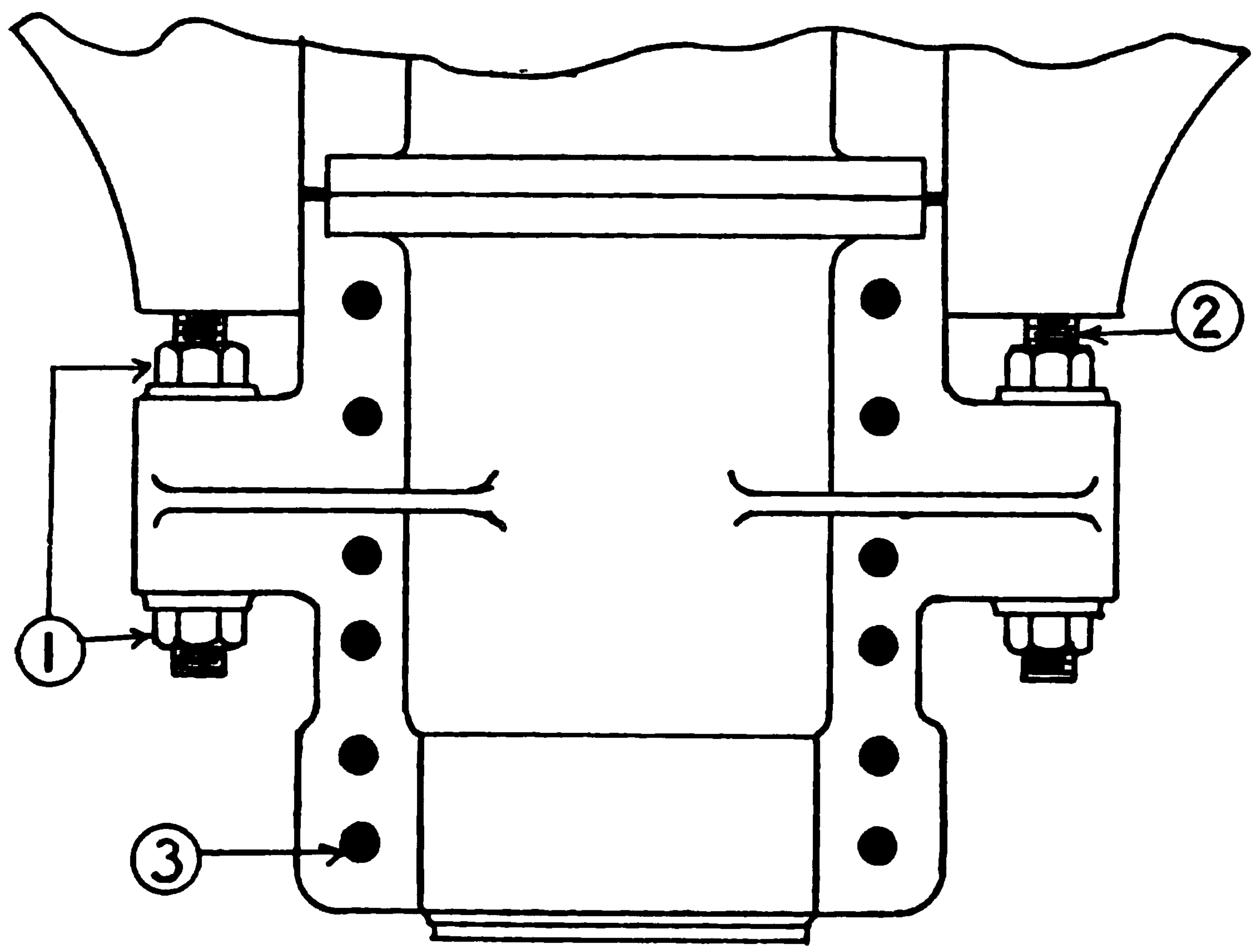
1. Guides for Cover. 2. Bolt holes, larger than bolts to allow for adjustment.

The reason for not boring the two end rings is to prevent the from having a too free escape. The thrust bush consists of two par top and bottom, the rings in the bottom bush bearing against t forward side of the collars on the shaft, and the rings in the top bu bearing against the after side of the collars on the shaft. The colla on the shaft are usually $1\frac{1}{8}$ in. apart, and the rings in the bush a 1 in. thick, so that there is a space of $\frac{1}{8}$ in. between the forward and a side of the collars. The rings on the thrust bush are bedded up the collars on the shaft, so as to ensure that all the rings are bearin Some firms grind the adjusting bush up against the collars on t shaft by means of ground emery and oil. When the rotor dumn is set to the requisite clearance, the bottom half of the thrust bush



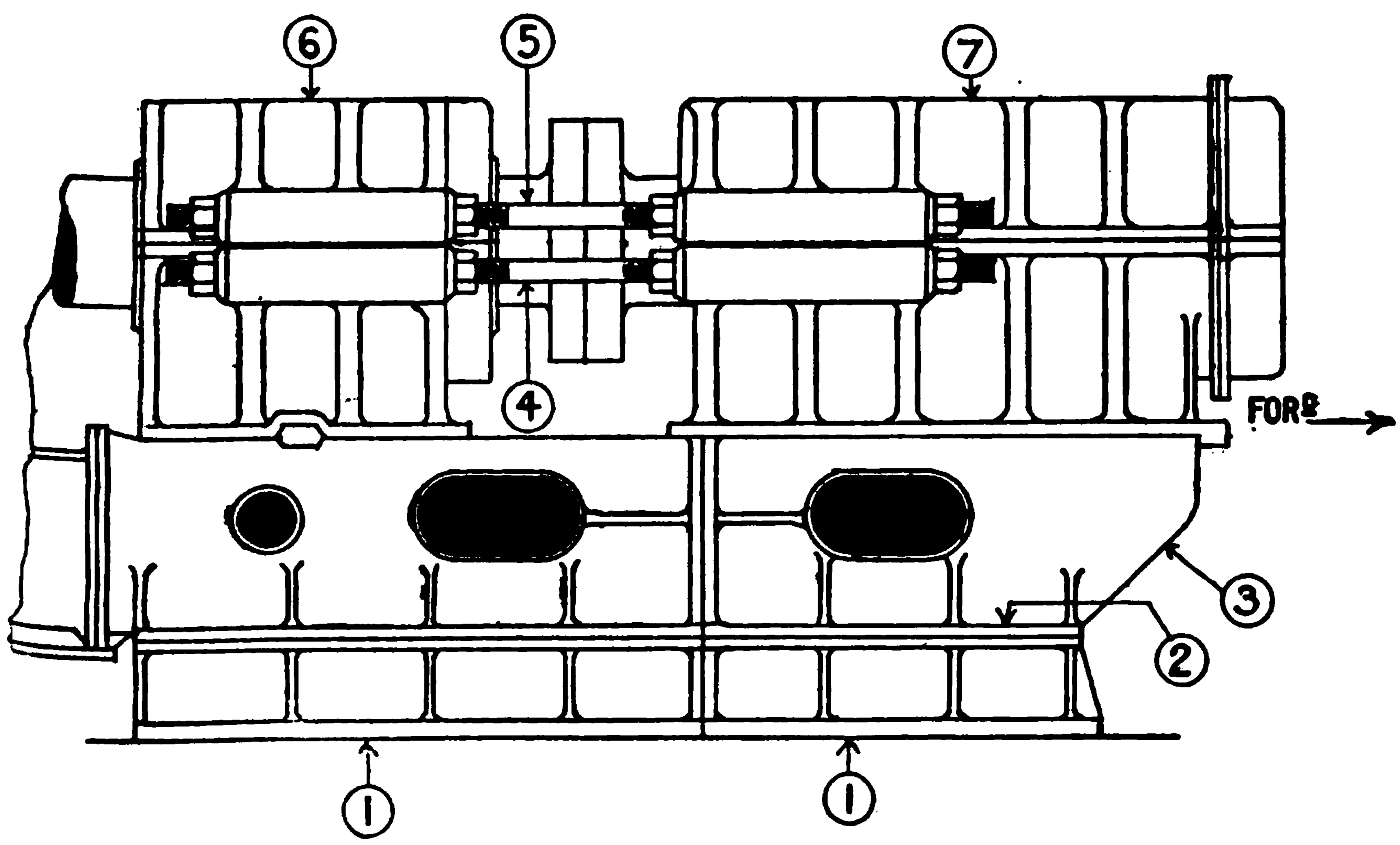
Lower Half for Ahe

(1) Ahead Thrust.
(2) Astern Thrust.
(3) Taper key for adjustment
(4) Taper key for adjustment



Method of Thrust Adjustment.

1. Adjusting nuts. 2. Adjusting studs bearing against face on casing end. 3. Bolt holes, larger than bolts to allow of adjustment.



Main Bearing, Adjusting Gear, and Thrust Seating of Large Turbines.

1. Rigid seating. 2. Bolt holes, larger than bolts to allow of expansion. 3. Sliding seat of thrust. 4. Adjusting studs for lower half thrust. 5. Adjusting studs for upper half thrust. 6. Main bearing cover. 7. Thrust cover.

put hard up against the forward side of the shaft collars, and in a space at forward side of the check on the outside of the bush a brass liner is fitted. The top cover, with bush in place, is then put on and brought forward so that rings in top half bear against aft side of shaft collars. It is then pushed or drawn aft until there is a clearance between the rings in top half of bush and collars on shaft of .004 in. to .015 in. This clearance is to allow for lubrication, and also for expansion of shaft when heated up. In average size turbines the gear usually fitted for moving the thrust bush consists of a screw fitted into a brass nut on the cover. The point of the screw spindle bears against a boss on the top half of the casing. By screwing up spindle, top half of bush may be moved forward, but the bottom half of bush can only be moved forward or aft by reducing or increasing the liner thickness in forward side of check on bush (see sketch).

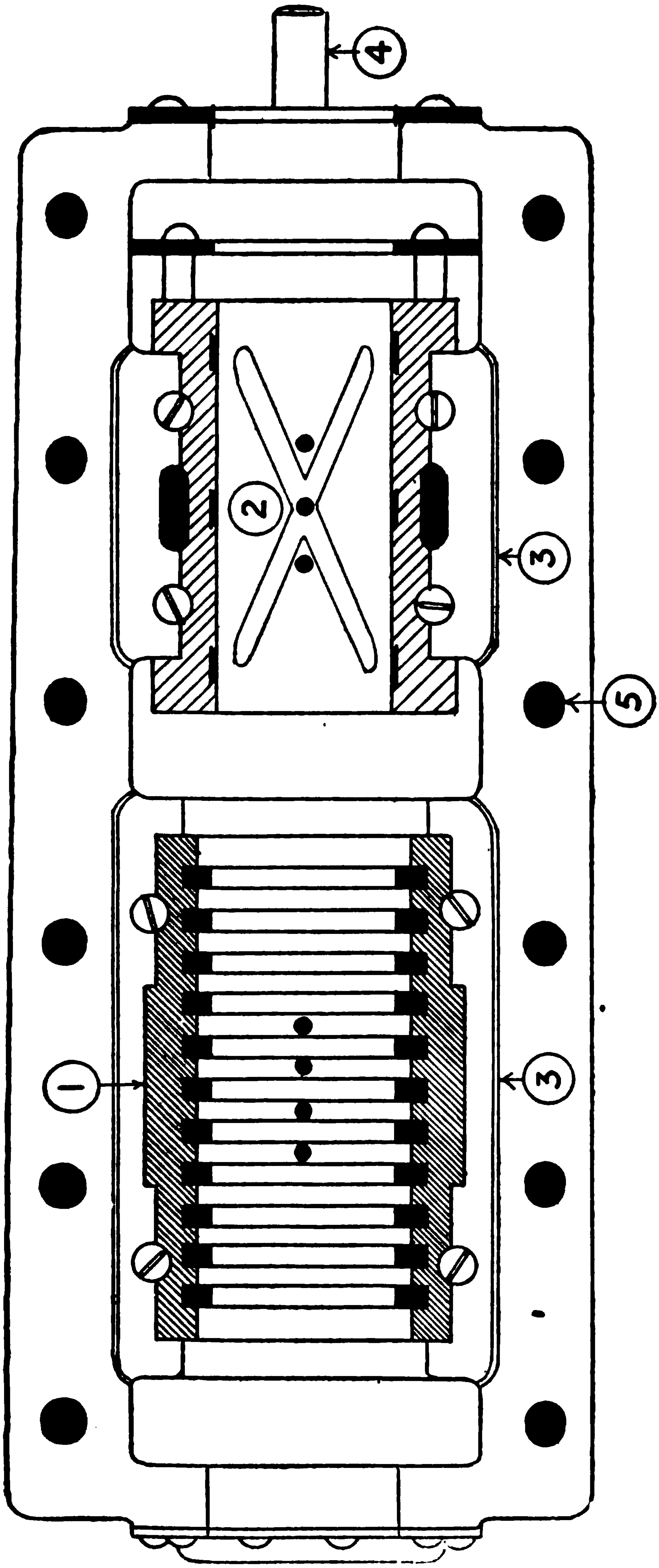
It is interesting to note that the wear on the block is practically *nil*, as the propeller thrust is balanced by the pressure of steam on the

Forward End View of Thrust Block.
1. Oil Holes in Brass Thrust Ring. 2. Adjustment Half Ring.
3. Shaft Collar.

vanes acting in the opposite direction, so that to all intents and purposes the block is not really called upon to receive the thrust as usually understood in connection with ordinary marine engines. The block is chiefly required to take the thrust when steam is turned on or off.

NOTE.—Sometimes the wear takes place *aft* instead of *forward* (usually in the H.P.) owing to the blade steam thrust being in that direction.

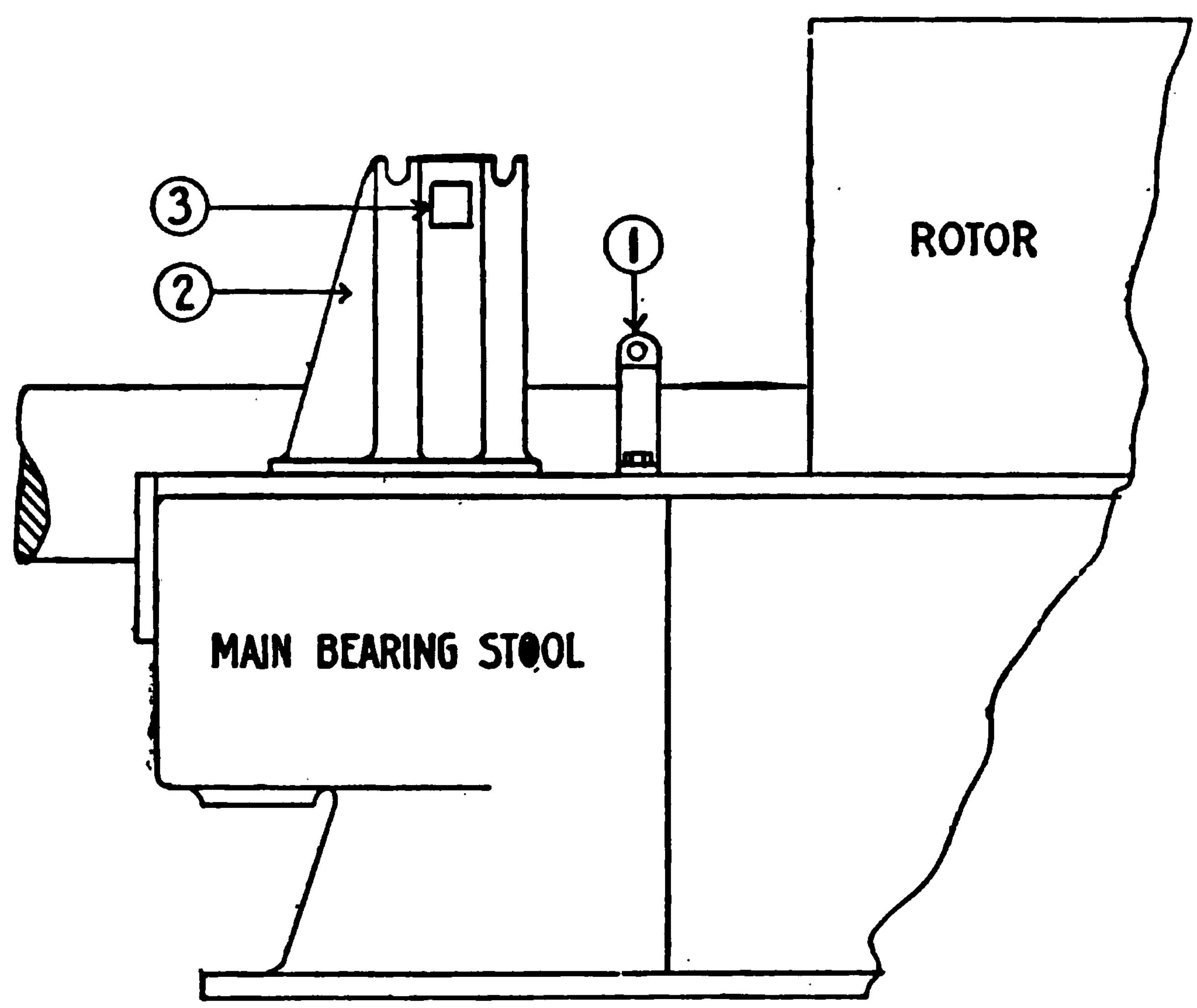
To adjust Dummy Clearance.—(1) By means of the dogs and "screwing-up" bolt forward, bring the rotor up until the dummy rings and grooves are in actual contact; next measure and note the clearance in thousandths between the after side of the finger plate and the spindle groove when in this position, and screw the rotor back until the clearance between the spindle groove and finger plate is *increased* by say $\frac{30}{1000}$ of an inch, or .03. The lower half or ahead section may then be locked in position by fitting in at the forward end or at both ends the adjusting rings.



Inside View of Thrust and Main Bearing Cover.

1. Astern Half Thrust. Oil Holes in Brass. 3. Oil Gutters. 4. Adjustment Stud. 5. Oval Holes to allow of adjustment.

(2) The upper or astern portion of the thrust is set as follows:—Ease back the nuts of the cover studs, and screw round the adjusting bolt in the cover until the rings and collars come into metallic contact by the cover moving forward. Now screw down tight the cover stud nuts, and adjust the stud until when tested by "feelers" the desired clearance of say $\frac{6}{1000}$, or .006, of an inch exists between the stud and the end of the casing; then ease back the cover nuts, and tap up aft the cover until the point of the adjusting stud and the turbine casing are in actual contact, which will thus give the determined thrust clearance; the cover bolts are then screwed down hard. It will be noted that the total clearance for oil is $\frac{6}{1000}$, or .006. The dummy clearance if when cold is, say, $\frac{30}{1000}$ of an inch, or .03, usually decreases to $\frac{20}{1000}$, or



Rotor Guide Brackets and Lifting Gear.

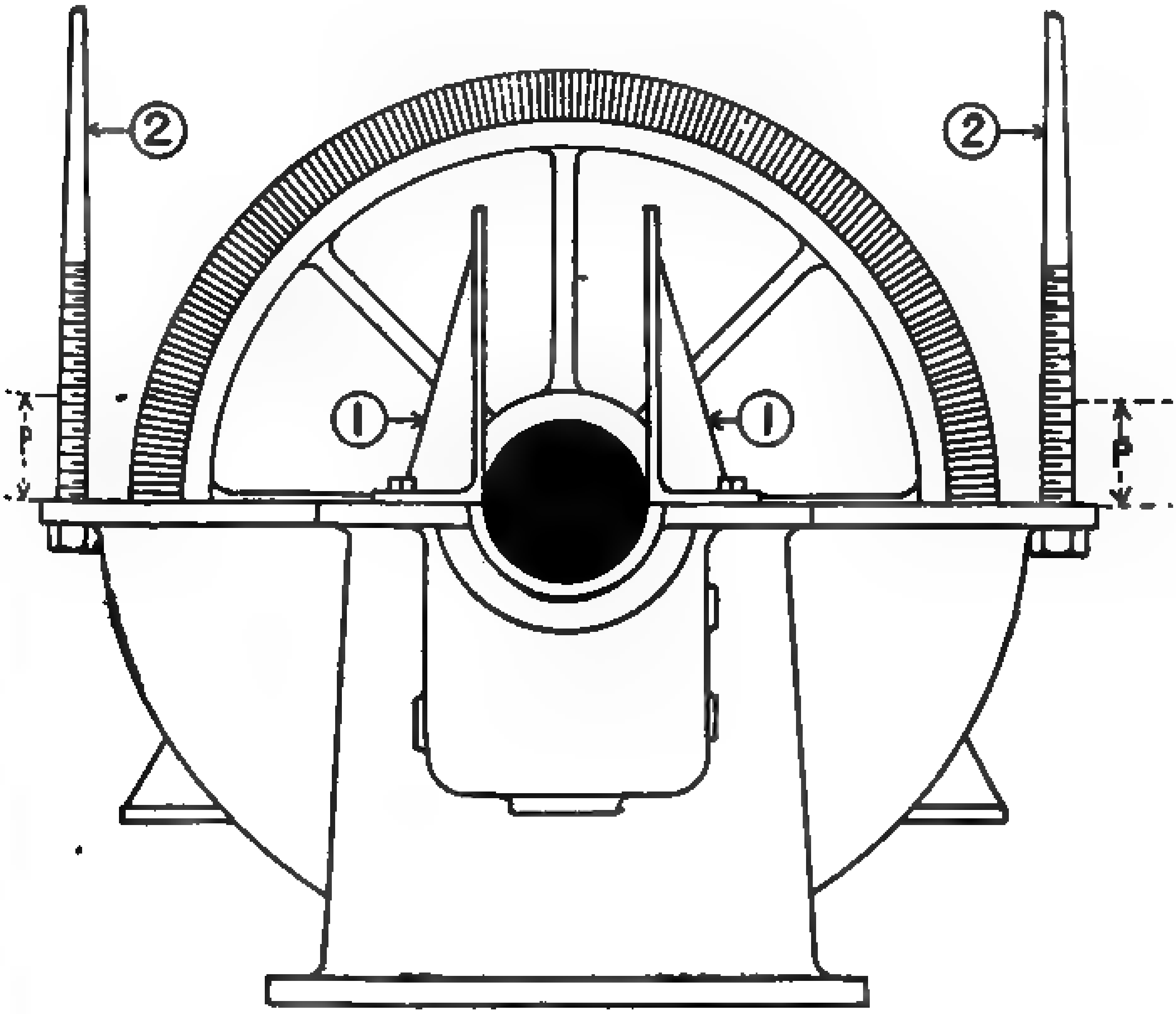
1. Lifting Strap. 2. Guide Bracket. 3. Slot for Carrying Bar.

.02, when expanded after heating up to working conditions. The engineers are therefore instructed to test the dummy clearance with the feeler gauge both previous to heating up and after heating up.

Some chief engineers test and note the clearance referred to every two or three days, or even oftener, this clearance constituting the most important and most delicate adjustment in the whole turbine, and one on which the economy and mechanical efficiency greatly depends. Should the dummy rings and grooves overstep the $\frac{20}{1000}$ or so clearance, breakdown is sure to occur by the stripping out of the small brass dummy rings; other damage may also result.

Bedding of Rotors.—The rotors being now balanced and blades tipped, the next operation is the bedding down of same. The rotors

are guided into position by means of guide columns at forward and aft end of turbine. These columns are set so that the rotor dummy will



End View of Turbine with Cover Lifted.

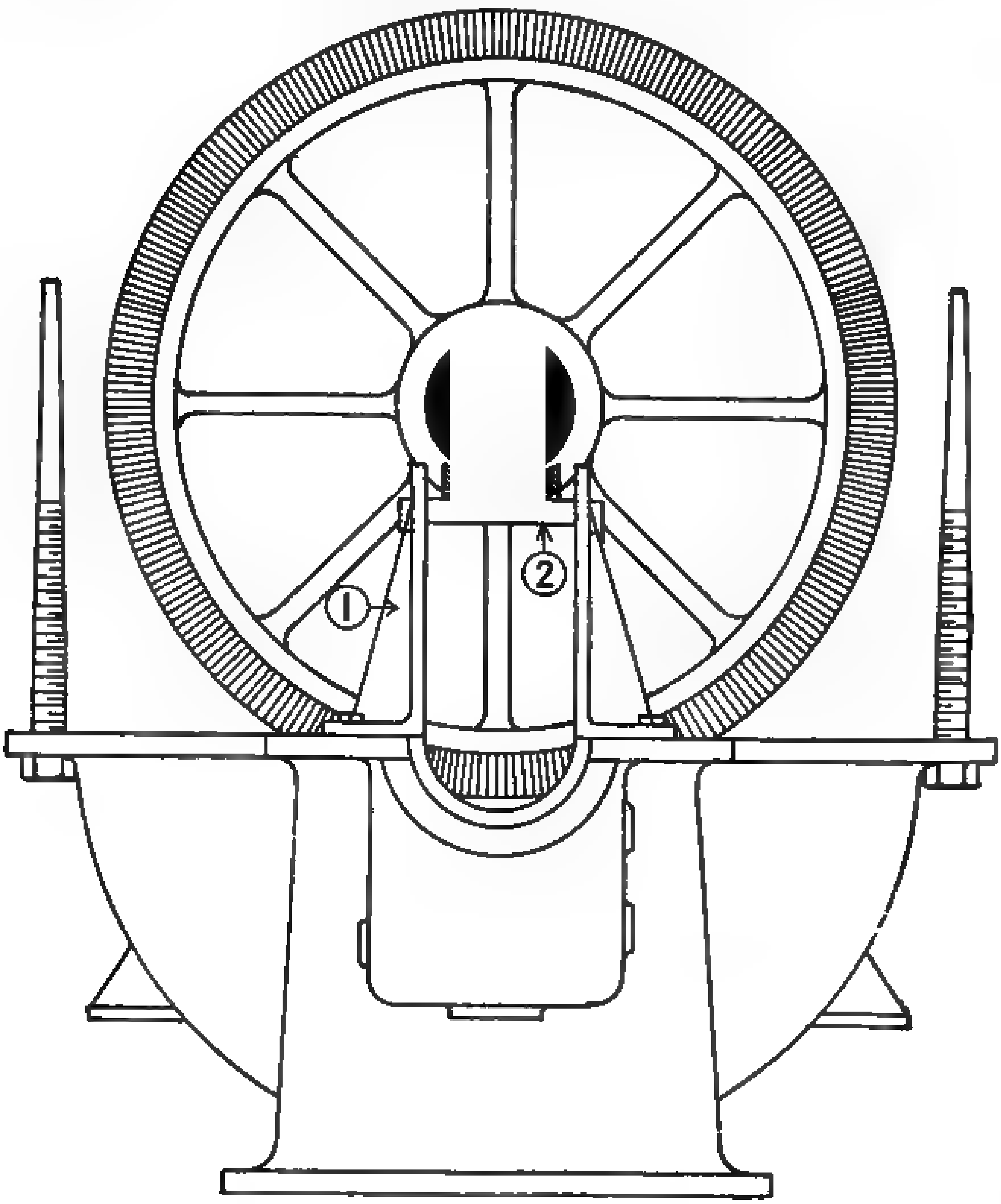
1. Guide Brackets for Raising of Rotor. 2. Guide Columns for Removal of Cover. P. Parallel Portion of Columns.

1. Strap for Lifting Rotor.

be about $\frac{1}{16}$ in. clear of blades in casing dummy. In the bottom of the casing between the rows of blades in each expansion melted wax is poured to a little more than what the blade clearance will be.

The rotor is revolved and cuts away any superfluous wax, leaving an amount equal to the clearance between the tip of the blades and the casing ; the blades are also gauged at the sides at the same time.

When the rotor is lifted the wax is taken out and gauged by



Rotor Clear of Casing and Resting on Carrying Bars.
1. Guide and Support Brackets. 2. "Carrying Bar" for Rotor.

micrometer. The bearings are now eased so as to ensure the requisite clearance on the sides, and wax is put on the rotor between the blades and also on the top half of the casing, which is put on, with the rotor in place. The rotor is revolved, and top half of casing lifted off and

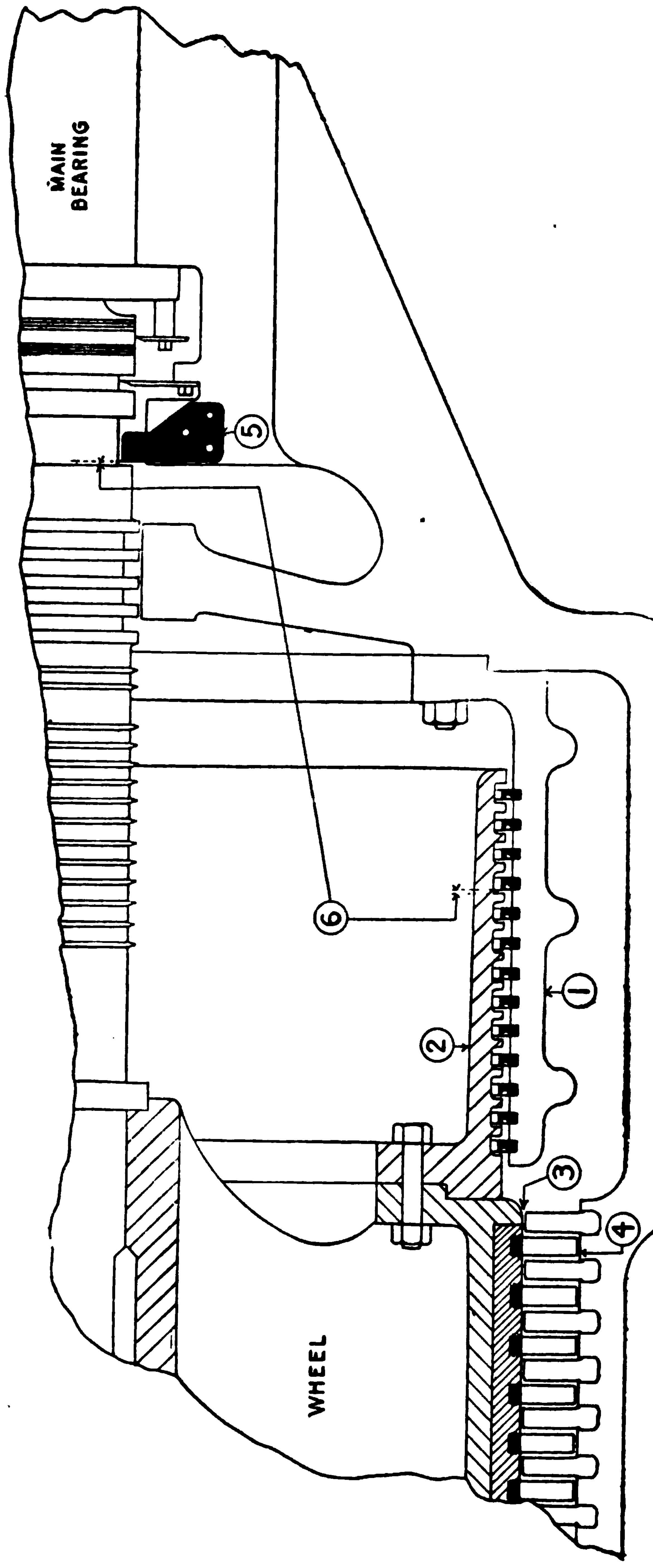
the wax gauged, which gives the top and bottom clearance of the rotor blades, and also the clearance of the casing blades. This having been adjusted to what is required, the dummy is now ground up. This is done by forcing the rotor up against the dummy blades while revolving, and ground emery and oil being poured in until all the blades on the dummy are at point of touch.

A gauge is now made showing the position of the rotor fore and aft. The rotor is put aft and the top half of the casing put on and bolted, and the dummy again brought into contact with the dummy blades and ground up on the top half until the rotor is at the same position as when the bottom half was ground, as ascertained by gauge. The thrust block is then adjusted and the longitudinal clearance of the rotor blades taken, also the radial and dummy clearance. The top main bearing bushes are leaded and the rotor lifted out, and casing finally cleaned and prepared for steaming to ascertain if the dynamic balance is correct. Guide columns are fitted for guiding casing and rotor into position, two at forward and two at aft end.

Steaming of Turbines.—In heating up turbines for steaming purposes, if the turbines are of large dimensions, it is usual to have heating steam pipes led into the casing at the forward and aft end. This ensures an equal heating up of the turbine.

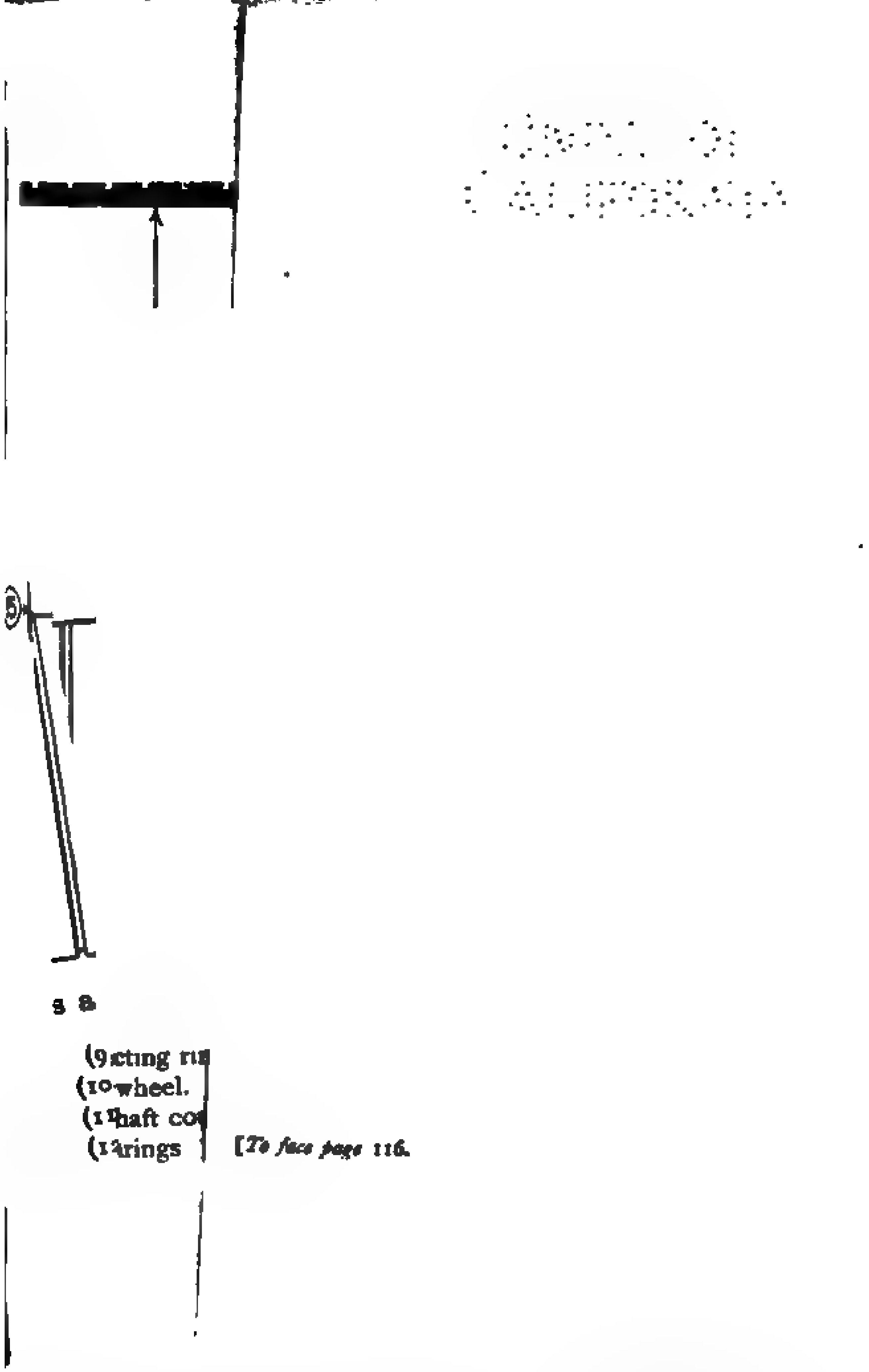
In small or average size turbines there are no heating pipes fitted, the steam being admitted at the forward end and passing aft. There are different opinions regarding which is the best way to heat up, some authorities preferring to heat up the turbine as quickly as possible so as to ensure an equal expansion all over, and others to heat up gradually. Steam being admitted into casing, the rotor should be turned about every two or three minutes, as if this is neglected the steam entering on one side of the casing will heat up one side of the rotor excessively, causing it to expand on that side, and thus alter the alignment of the shaft and cause it to bind in the bearings. While heating, attention should be paid to the finger piece.

When the dummy is set to the required clearance the "finger piece" is also set to the same, having a clearance from the aft side of the groove on shaft equal to the clearance of the dummy. This should be gauged by feelers while heating up, more especially if for the first time, and should the dummy clearance decrease until the clearance is only .01, steam should be shut off so as to allow what heat has been imparted to transmit itself through the body of the wheels to the shaft. The shaft being expanded against the thrust will thus cause the rotor to expand in an aft direction. The above explanation is only applicable to steaming turbines in the shops, where no steam connections to the glands are fitted. The oil should be circulated through bearings for two or three hours before starting, and the filter opened and cloths cleaned so as to ensure that all grit and dirt has



Dummy and Finger Plate showing how Clearance is Measured.

1. Dummy Casing.
2. Dummy Piston.
3 Steam Admission to Blades.
4. 1st Expansion Blades.
5. Finger Plate.
6. Dummy Clearance and Finger Plate Clearance Equal.



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kept running while the turbines are stopped temporarily. This ensures the withdrawal of any condensed water from the turbine casings. To prevent air finding its way back into the L.P. turbine casings, a U bend is arranged on the air pump suction pipe to act as a "water seal," and as this bend contains water, the admission of air is prevented even should the air pumps cease working. The non-return

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2. Du
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been washed out of bearings. All drain connections should be left open, and it is preferable to have an ejector fitted and drains led away so that they can be left open while running, as owing to length of piping required to transmit steam to turbines, in most shops, there is bound to be a large amount of condensation. As before mentioned, when balancing, the turbines should be run up to 20 per cent. above their working speed to ensure a good dynamic balance.

When running a turbine in the shop under steam up to the required speed, after the steam is shut off the rotor usually continues revolving for some minutes before coming to a dead stop, thus proving the correctness of the balance.

In the case of large turbine rotors after steaming the drums have been found slightly out of truth due to unequal expansion; this necessitated further turning up in the machines, and in some cases filing up of some of the blade rows at certain positions to maintain the required tip clearance.

It has also been discovered that the blade tip side clearance is affected by the weight of the upper half casing, as in one case tested the side clearance measured $\frac{15}{1000}$ in. with the upper half cover off and $\frac{60}{1000}$ in. with the cover bolted down. This is understood to be due to the weight of the upper casing producing a "spring" on the lower half and thus opening up the cylinder diameter horizontally, with resulting increase of blade tip side clearance.

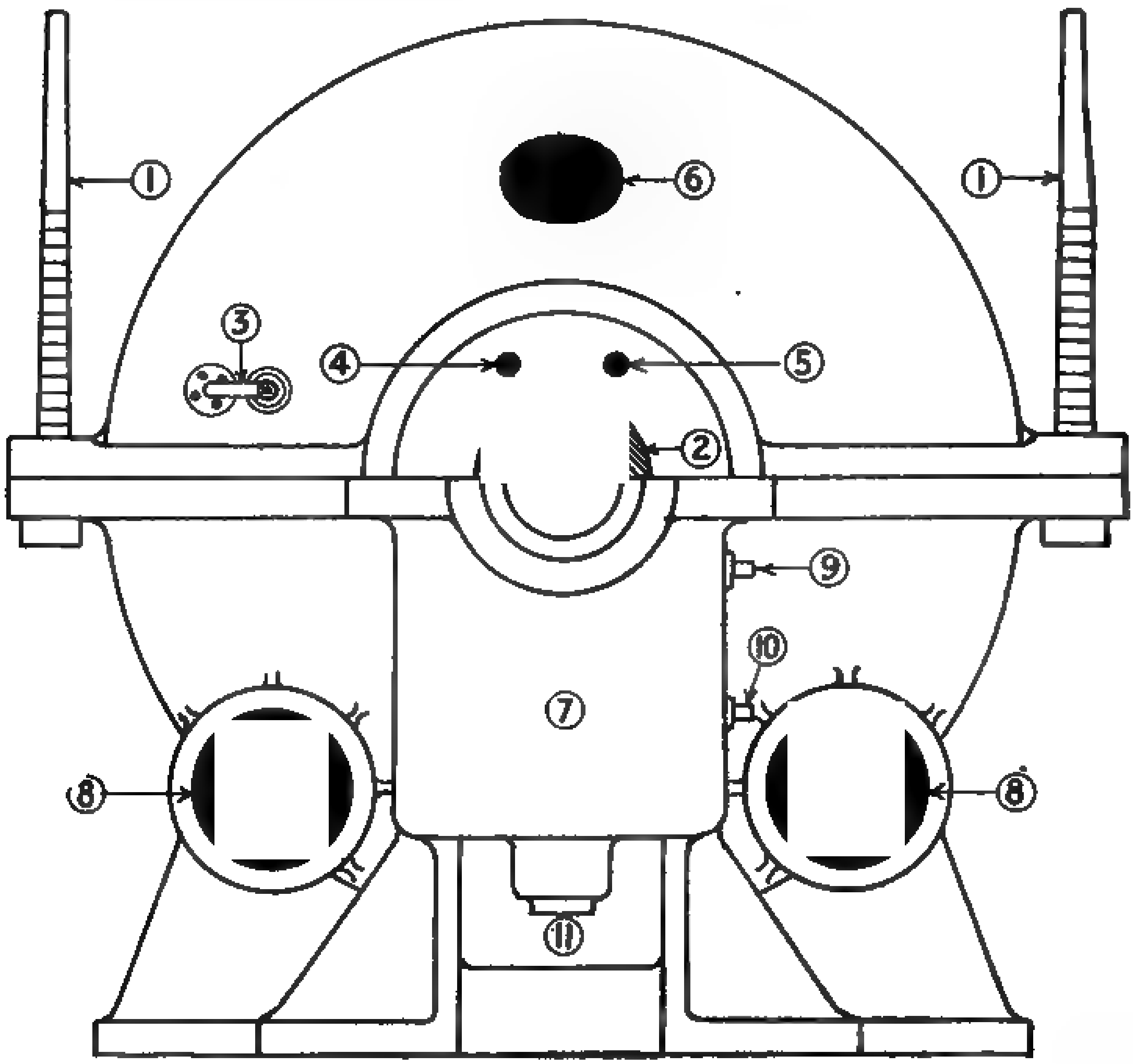
Guide Columns.—Four long guide columns are screwed into the lower half turbine casing, and these are intended to maintain the upper half casing in its correct position when being raised or lowered, and thus prevent possible damage to the blades by contact. These columns are placed two at each end, and they are carefully marked in inches, and parts of inches, to allow of the cover being raised or lowered evenly throughout the length of the turbine, and thus obviate canting.

Turbine Casing Drains.—A drain pipe and cock or screwdown valve connects the dummy casing at the forward end of each turbine casing with one of the last expansions aft. This allows any condensed water to drain out from the higher forward position of the casing to the lower after position. These drain connections are only kept open when the turbines are stopped temporarily.

L.P. Turbines.—Large double drain pipes and specially light non-return valves are fitted at the after end of each L.P. turbine, and these are connected to the "wet" or ordinary air pumps which are kept running while the turbines are stopped temporarily. This ensures the withdrawal of any condensed water from the turbine casings. To prevent air finding its way back into the L.P. turbine casings, a U bend is arranged on the air pump suction pipe to act as a "water seal," and as this bend contains water, the admission of air is prevented even should the air pumps cease working. The non-return

valve mentioned above would prevent the possible return of water back from the pump into the turbine casings.

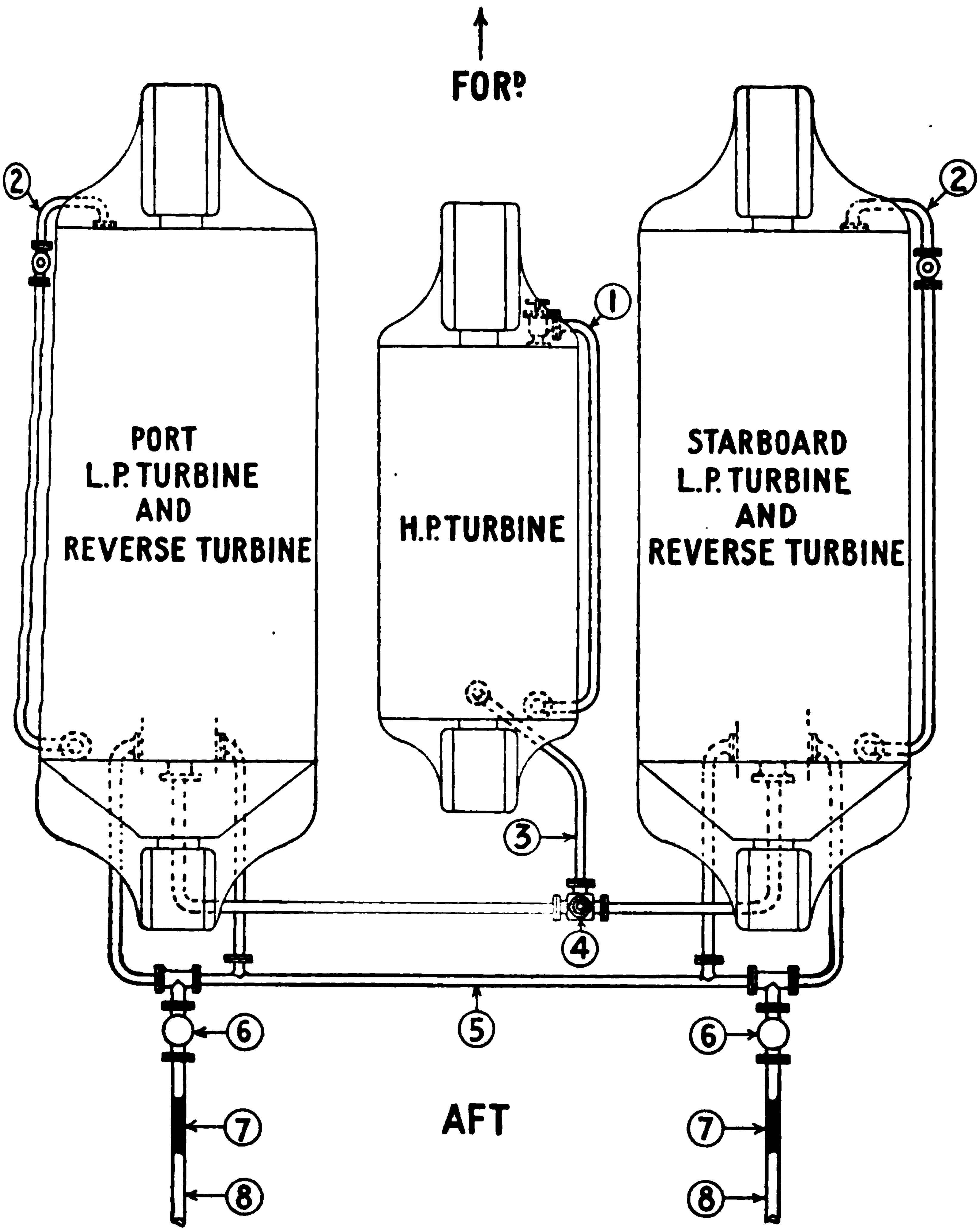
H.P. Turbine.—The drain from the after end of the H.P. turbine is merely led to the drain pipe connection of either of the L.P. turbines, a cross or "change over" connection to either L.P. being arranged by means of a two-way cock.



End View of H.P. Turbine (Thrust Cover removed).

1, 1. Guide Studs. 2. Rotor Spindle. 3. Rotor Clearance Gauge. 4. Gland Steam Inl 5. Gland Steam "Leak-off." 6. Hand Hole. 7. Thrust Seating. 8, 8. Steam Inl to Turbine. 9. Oil Supply. 10. Cooling Water Inlet. 11. Oil Outlet.

NOTE.—The water which is constantly drained off from the L.P. turbines when running is formed by the adiabatic expansion of the steam in the turbines. The H.P. exhaust pipes to each L.P. turbine are placed low down to allow the water condensed in the H.P. turbine to pass easily to the L.P. turbines, and afterwards be drained off by the connections aft to the "wet" air pumps. (See sketch facing page 120.)



Turbine Drain Connections.

1. Drain from forward to after end of H. P. Turbine.
2. Drain from forward to after end of L. P. Turbines.
3. Drain from after end of H. P. Turbine to after end of either L. P. Turbine.
4. Two-way Cock for changing over.
5. Drain from after end of both L. P. Turbines to "Wet" Air Pump.
6. Light Non-return Valves.
7. *U* Bend in Pipe to act as "Water Seal" and prevent return of Air.
8. "Wet" Air Pump Suction Pipes.

Note.—The connections marked 1, 2, and 3 are only open when the turbines are pped, and the connections marked 5, 6, 7, and 8 are open when the turbines are either ning or stopped (temporarily).

All casing drains can be opened up to the bilges when requir and this is done when the engines are rung off, so as to get rid of t vacuum existing in the turbine casings.

The two drains led from the H.P. exhaust pipes to the L casings aft are often just kept eased off the face when running, so to allow the condensed water only to be drained off, while in ot cases these are shut altogether when running.

Rotor Screw.—When the rotor requires to be moved lo tudinally, a pin is screwed into the forward end of the spindle, wh is tapped out for the purpose, and by means of a dog arrangem and double nuts the rotor can be moved forward or aft as requi It is sometimes necessary to disconnect the first length of shaft when screwing up the rotor as described.

Steam Connections.

The usual steam connections on the rotor casing are as follow

Ahead Turbines.

1. **Steam Inlet at Forward End.**—In the H.P. turbine th direct from the boilers, and in the L.P. turbines from the H.P. exh

2. **Exhaust at After End.**—In the H.P. turbine this leads t two L.P. turbines as initial steam, and in the L.P. turbines dire the condenser.

3. **L.P. Turbine Non-return Valves.**—These valves, arrang prevent the return of steam from the L.P. to the H.P. turbine fitted in a chest on the L.P. casing where the H.P. exhaust br connects, and being loaded by means of two springs, only allow c admission of steam from H.P. to L.P. casings, but close immedi steam attempts to pass back to the H.P. from the L.P. These v come into action when high-pressure steam is supplied direct t ahead L.P. turbines in manœuvring or working with the outside s only.

4. **Direct Steam to L.P. Turbine.**—This supplies high-pre steam direct to the ahead L.P. turbines, and is required when run ahead with the outside shafts only.

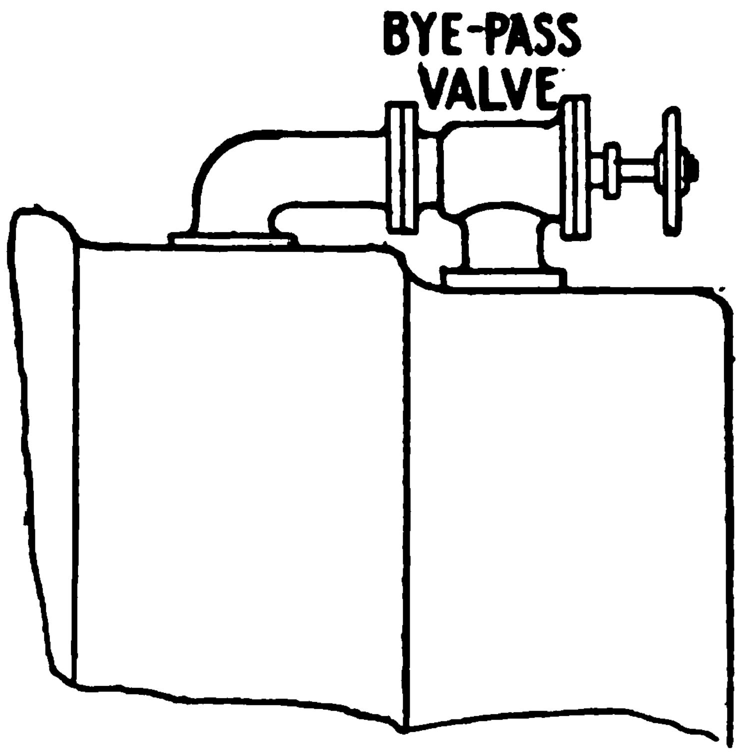
5. **Bye-Pass Steam.**—This consists of either one or two h valves and pipe connections leading from the 1st to the 3rd expa of the H.P. turbine to admit high-pressure steam direct to th expansion if required to increase the power, or to create an equal pressure in starting up and prevent possible damage to the blad excessive vibration.

6. **Escape or Relief Valves.**—Escape valves, loaded to a su pressure, are fitted usually on the top of the two L.P. turbine ca forward and aft, and one on the H.P. turbine casing aft.

7. **H.P. Gland Steam.**—Steam of reduced pressure is admitted to the H.P. gland pockets only when heating up to allow of gradual expansion of the metal; but when running this connection forms a "leak-off" to one of the L.P. turbines; this is arranged for by means of a two-way cock.

8. **L.P. Gland Steam.** — Both ends of each L.P. turbine have low-pressure steam admitted to the gland pocket to prevent the admission of air. No "leak-off" is fitted, and the pressure of steam varies from 1 to 4 lbs.

In both the H.P. and L.P. turbines the exhaust steam from the last ring of blades flows into the *inside* of the drums, and the steam glands are therefore required to pack the spindle against outflow of steam in the case of the H.P. turbine, and against the admission of air in the case of the L.P. and reverse turbines. The pressure inside of the H.P. rotor drum is usually somewhat between 10 and 25 lbs., and



Bye-pass from 1st to 3rd Expansion (on H.P. Turbine only).

the vacuum inside the L.P. rotor drum approximates to that carried in the condenser, being probably between 2 or 3 lbs. less. As before stated, the H.P., L.P., and reverse dummies act as packing to prevent the outflow of the higher or admission pressure steam from the turbine casing into the atmosphere; whereas the steam glands on the rotor spindles are only required to pack the turbine against the exhaust or lower pressure steam.

9. **Dummy "Leak-off."**—When a dummy leak-off is fitted, two pipes are usually led from the ahead dummy casings to the 3rd expansion of the turbine, where the pressure is less, to take away any steam which may leak through the first series of dummy rings and grooves.

Reverse Turbine.

10. **Direct Steam.**—The reverse turbines only require one connection, that of the direct steam at the after end, which, after expand-

ing through the reverse turbine blades forwards, exhausts into the condenser by the common exhaust branch. Small drain holes are fitted in the bottom of the reverse turbine casing.

Steam and Reverse Valves.—Steam can be admitted by the large hand-controlled valve to the H.P. turbine direct, and so through the series of turbines, so that all are working at once. Two smaller hand-valves also admit full-pressure steam to a pair of chests, each containing a piston valve which is actuated by patent steam and hydraulic gear. Round ports are cast in the chest, top and bottom, one end admitting steam to the ahead L.P. turbine, and the other admitting steam to the reverse L.P. turbine, and the valve is moved up and down by the gear to uncover the ports and admit steam from the centre of the piston valve as required, the steam at the same time being admitted to the chest itself by means of the large hand-valve referred to.

This constitutes the reversing and manœuvring gear.

Steam Glands.

The steam glands on turbines differ greatly from the steam glands on the reciprocating engine. In small turbines the usual type of gland consists of a series of brass ramsbottom or split rings fitted into grooves in the shaft, over which a sleeve of hard grained cast iron is fitted. The rings are usually of phosphor bronze. They are turned to a larger diameter than the inside of the gland, then so much is cut out, so as to give an opening at the joint of $\frac{1}{32}$ in. to $\frac{1}{16}$ in. when rings are in place inside of sleeve. This opening varies according to size of ring and the expansion of same when heated. Another style of gland, and one being largely adopted, is a combination of the ramsbottom ring and brass fin blades similar to those fitted in the radial or astern dummy. This form is usually termed labyrinth packing. The brass blades are fitted into grooves in the rotor spindles and caulked into place; they are then knife edged, and have a radial clearance at the tip of .010 in. to .020 in. The gland or sleeve enclosing the same is in two halves, and is of hard grained cast iron, grooved and bladed similar to the spindle. The blades in the spindle fit in between the blades in the sleeve. On the outer end of the shaft there are usually four brass ramsbottom rings which bear against another sleeve which is bolted to end of the casing (see sketch). Between the labyrinth packing and the ramsbottom rings a space is left which forms a pocket or receiver, and from this point the gland escape or leak-off takes place. This in an H.P. turbine is led by piping to the 2nd or 3rd expansion of the low-pressure turbine. The principle of the labyrinth packing is the same as in the radial dummy. The steam is wire-drawn in passing the tip of the blades, and then expanded into the space between the blades, and so on until it reaches the escape or leak-off pocket. Any leak-off that takes place past the four ramsbottom rings is also collected, and led away by a vapour

pipe either to the atmosphere or to the hot-well tank. In the case of a low-pressure gland, where the interior of the turbine is in a vacuum, the admission of air must be prevented: to do this, steam is admitted into the pocket between the ramsbottom rings and the labyrinth packing at a pressure from 1½ to 3 lbs. per sq. in., which low pressure is sufficient to prevent the admission of air

Steam Gland (Naval Type).

1. Leak-off Ports (Inner).
2. Leak-off Ports (Outer).
3. Leak-off Pipe Connection.
4. Leak-off Pipe Connection.
5. Oil to Gland Rings.
6. Oil Baffle Plate.

which would otherwise affect the vacuum. On the high-pressure gland there is also a steam inlet connection fitted to the leak-off pocket, and this connection is used when heating up the turbine preparatory to running: this ensures the expansion of the shaft in relation to the casing and rotor. By opening steam on the forward end and heating up the shaft, the shaft will expand and press against thrust rings, and so force rotor aft, and increase or keep the dummy clearance at the

Steam Gland.

(For large-sized Turbines.)

1. Ramsbottom Rings.
2. Radial Rings.
3. "Leak-off" Pocket.
4. Pipe Connection.
5. Vapour Escape.
6. Oil to Gland Rings.
7. Baffle Plate.

Steam Gland.

(For average size Turbines.)

1. Ramsbottom Rings.
2. Radial Fin Rings.
3. "Leak-off" Pocket.
4. Baffle Plate.

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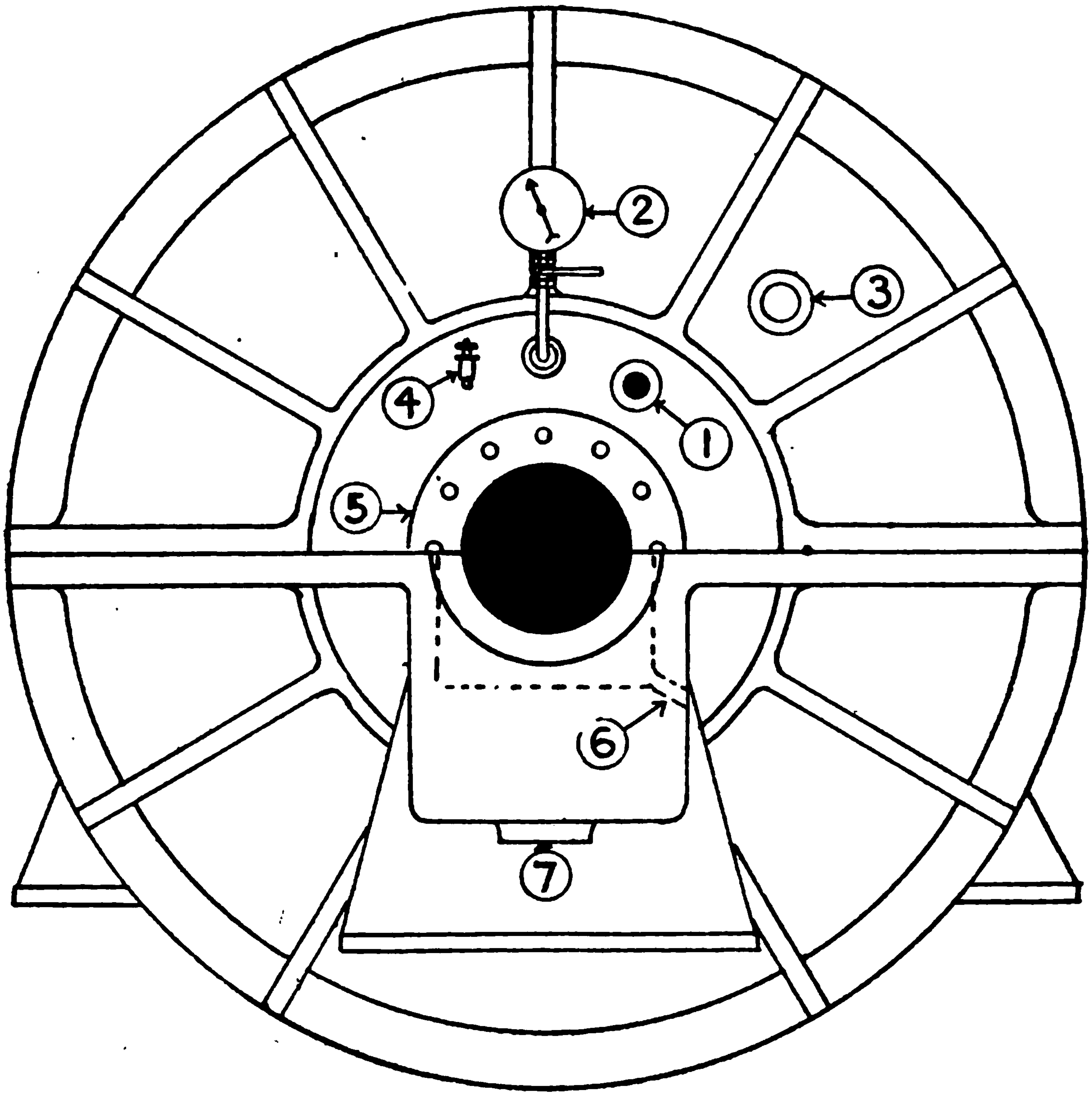
drain. On the steam glands at the leak-off and vapour pockets steam valves are fitted, also a connection admitting steam to the interior of the glands. If the exhaust pipe connection on the H.P. turbine is from the bottom half of the casing, a drain connection from each pipe is led to the bottom of after-end of low-pressure casing, with a shut-down valve between same. An extended spindle is fitted

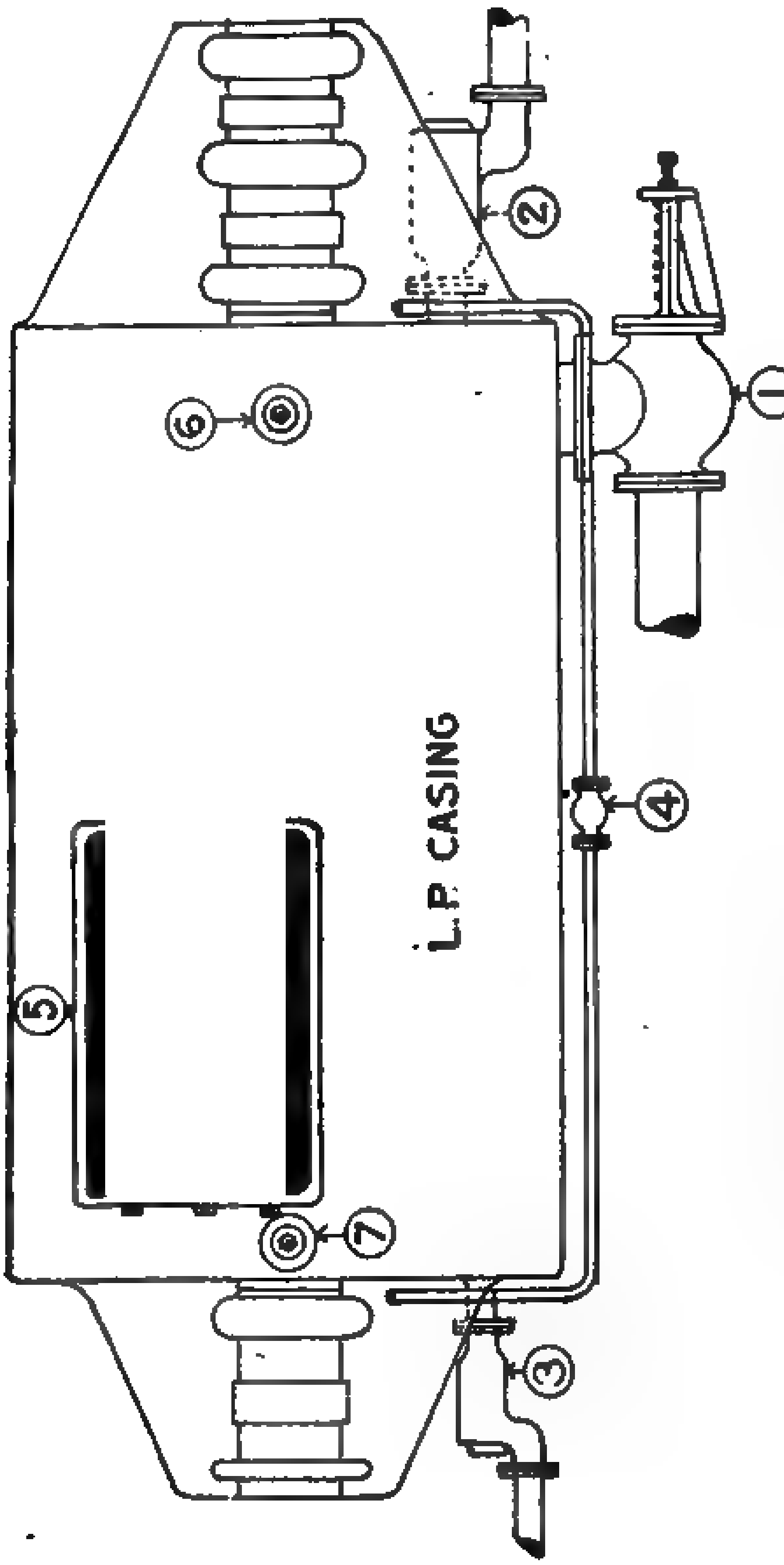


End View of H.P. Turbine.

1. Gland "Leak-off."
2. Pressure Gauge for Gland.
3. Micrometer Gauge.
4. Oil to Glands.
5. Gland Case.
6. Drain from Gland Pocket to Hot-well.
7. Oil Drain to Cooling Tank.

to these valves, and led up above the platforms so as to be easily worked. The after end of low-pressure turbines being in a vacuum, water which accumulates in the high-pressure turbine is drawn by the drain pipes into the L.P. casing. A connection is also made from bottom of low-pressure after end to air pump suction, so that in heating up turbines water can be drawn from both H.P. and L.P.'s to air pump and then into hot-well.

Starting and Reverse Steam Valves.—The steam in its passage from the boiler to the turbines passes through the boiler stop-valves, then through a master valve ; next past the throttle or governor valve.

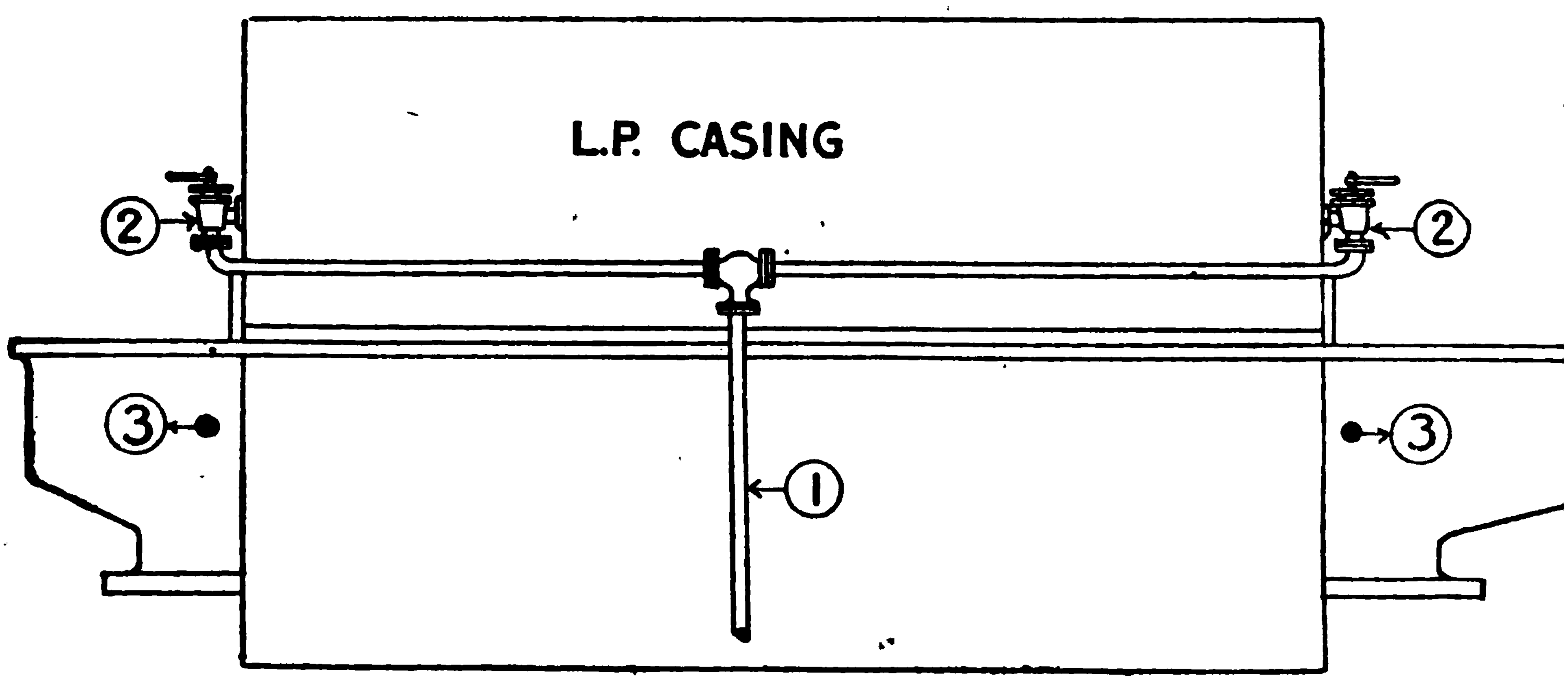


Plan of L.P. and Reverse Casing.

1. Steam Inlet to L.P. from H.P. Exhaust.
2. Ahead Steam direct from Boilers.
3. Astern Steam direct from Boilers.
4. Steam to Glands.
5. Exhaust to Condenser.
6. Relief Valve on Forward End.
7. Relief Valve on After End.

then the main stop-valve, from which the pipes to the high-pressure turbine are connected. After passing through the H.P. turbine the steam is led through two pipes, one to each L.P., and then it passes

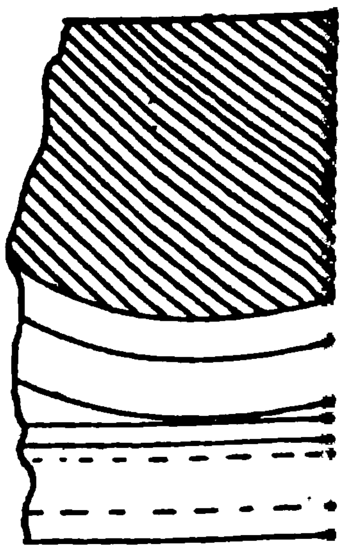
through the exhaust pipe into condensers. When manœuvring, that is, working the two outside shafts, steam is admitted to these manœuvring valves from a connection below the throttle valve into two chests, one for port and one for starboard. In these chests are two valves, one admitting steam to low-pressure turbines and the other steam to astern turbines. Governors are fitted sometimes, of the ball type, connected up to the throttle valve, this valve being placed between the boilers and the main stop-valve and manœuvring valves, should turbines race; then if throttle valve shuts, steam is cut off completely. In large turbines having the astern turbine separate from the L.P., a steam heating connection is fitted so as to keep this turbine warmed up, ready to start when required.



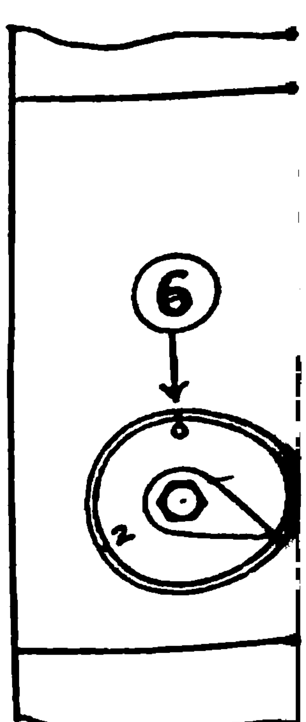
L.P. Casing Connections.

1. Steam to Glands. 2. Cock to admit Steam to Pocket of Gland. 3. Gland Drain to Hot-well.

Dummy Adjusting Gear.—For turbines of large power and weight, such as those fitted in battleships, cruisers, and large passenger steamers, the thrust block adjusting is usually arranged as shown in the sketches. After the lower and upper half blocks are set by the worm gear shown, to the required dummy clearance, the two are locked by means of wedge keys (one for each half block) which are screwed up by the nuts fitted on each end of the keys for that purpose. Indicators are fitted which register the amount of longitudinal movement given to either the lower (ahead) or upper (astern) half of the thrust block, which, of course, regulates the dummy clearance. Movement is given to each half block by a pair of horizontal shafts which are keyed to prevent rotation. On these shafts the worm wheels are screwed; when, therefore, the worm is revolved, the worm wheel of the shaft also revolves, and as this wheel is prevented from moving



End view



View of I

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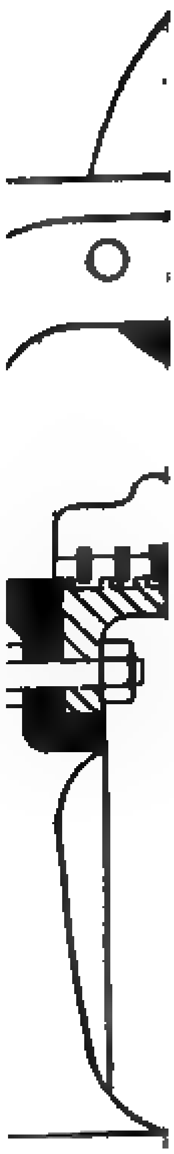


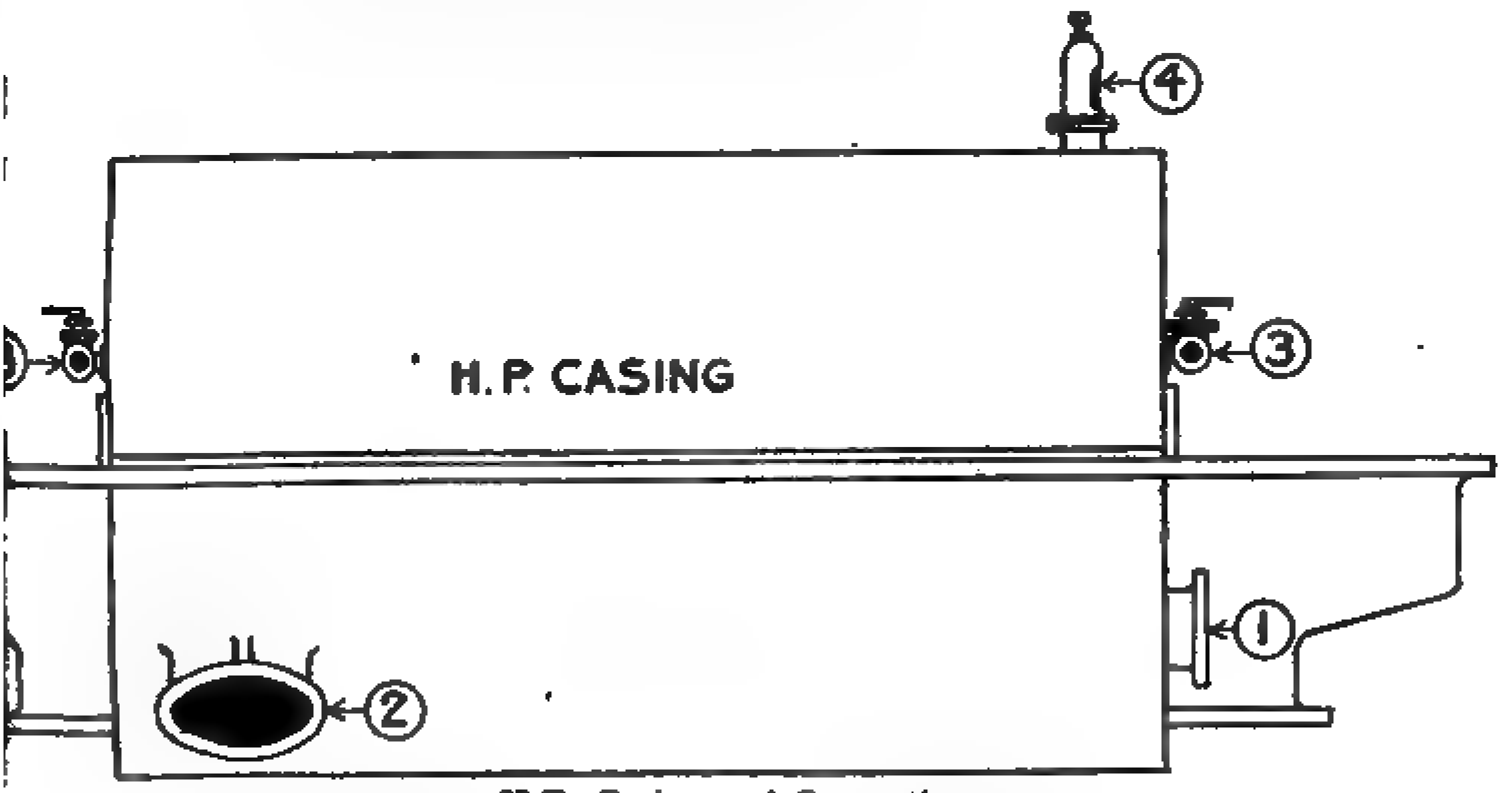
adjustment is obtained by means of the taper keys and worm gear described above, the instructions as to setting are as follows:—

(1) When the dummies are in contact and the top and bottom thrust collars hard up against thrust rings, the indicators read zero when nuts are tightened up.

(2) Each revolution of indicator nut is equal to .003 in. forward or aft movement of block.

(3) Before moving the top half of block aft, or the bottom half forward, the top or bottom wedges must be slackened away.

(4) When running, the top wedge indicator should read .010 more than bottom one to allow of oil clearance film.



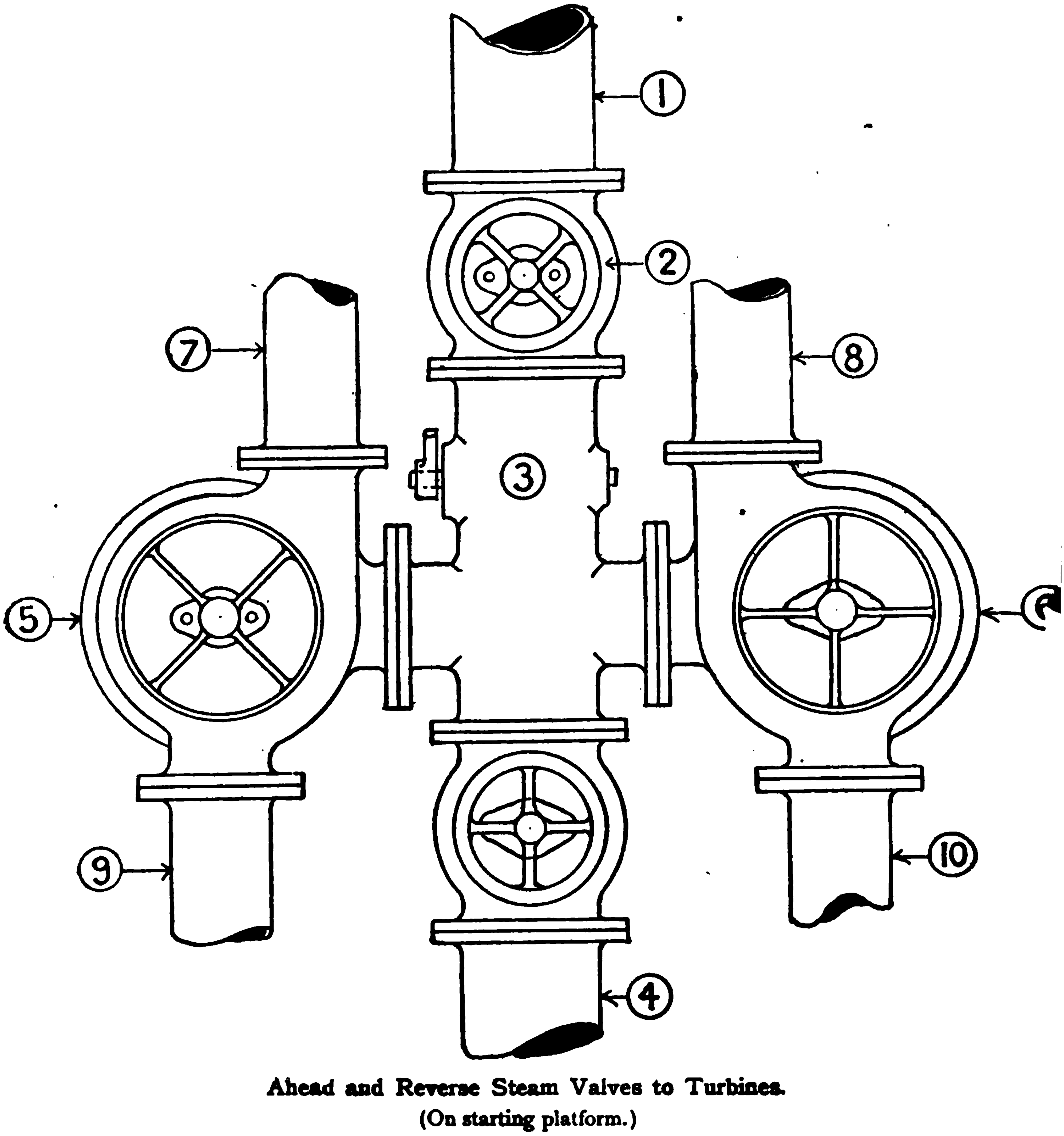
H.P. Casing and Connections.

1. Boiler Steam Inlet.
2. Exhaust Steam Outlet to L.P. Turbines.
3. Reduced Steam to Glands.
4. Relief Valve.

To Reduce Dummy Clearance.—Ease back bottom half block aft, slack indicator nut until it reads the required clearance, then screw up back nut to secure wedge. With bottom worm gear screw forward till it bears on wedge, then with top gear ard, following up with top wedge by easing away ightening up back nut until the top indicator gives s the bottom one, *plus .010;* finally screw top half bears hard on wedge.

ummy Clearance.—Screw bottom half of block with wedge by easing back nut and screwing up the required clearance is obtained, now screw up wedge, and screw bottom half block forward until it e. It should be observed that the thin end of the the thick end of the top wedge are on the same

side as the indicators; also, that in screwing the rotor either forward or aft it will require to be turned meanwhile.



Ahead and Reverse Steam Valves to Turbines.
(On starting platform.)

1. Steam from Boilers.
2. Master Valve.
3. Throttle Valve.
4. Steam to H.P. Turbine.
5. Manœuvring Valve for Port L.P. Turbine.
6. Manœuvring Valve for Starboard L.P. Turbine
7. Steam to Port Astern Turbine.
8. Steam to Starboard Astern Turbine.
9. Steam to Port L.P. Ahead Turbine.
10. Steam to Starboard L.P. Ahead Turbine.

Strainers.—On each of the steam inlet flanges on the turbines strainers are fitted, through which the steam passes on its way to the

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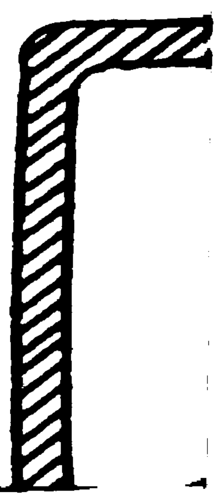

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shaft to the wood varies according to size of shaft. These arms are also levelled, and in front of them suitable platforms are arranged on which to work a surface gauge. The movable end of the shaft is

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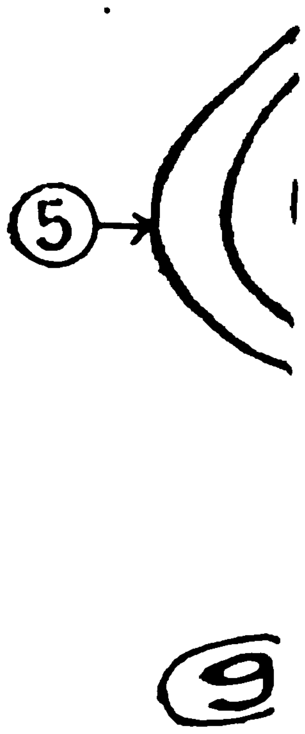


1. Steam
2. Maste
3. Throt
4. Steam
5. Manoeuvring Valve for Port L.P. Turbine.

10. Steam to Starboard L.P. Ahead Turbine.

Strainers.—On each of the steam inlet flanges on the turbines strainers are fitted, through which the steam passes on its way to the

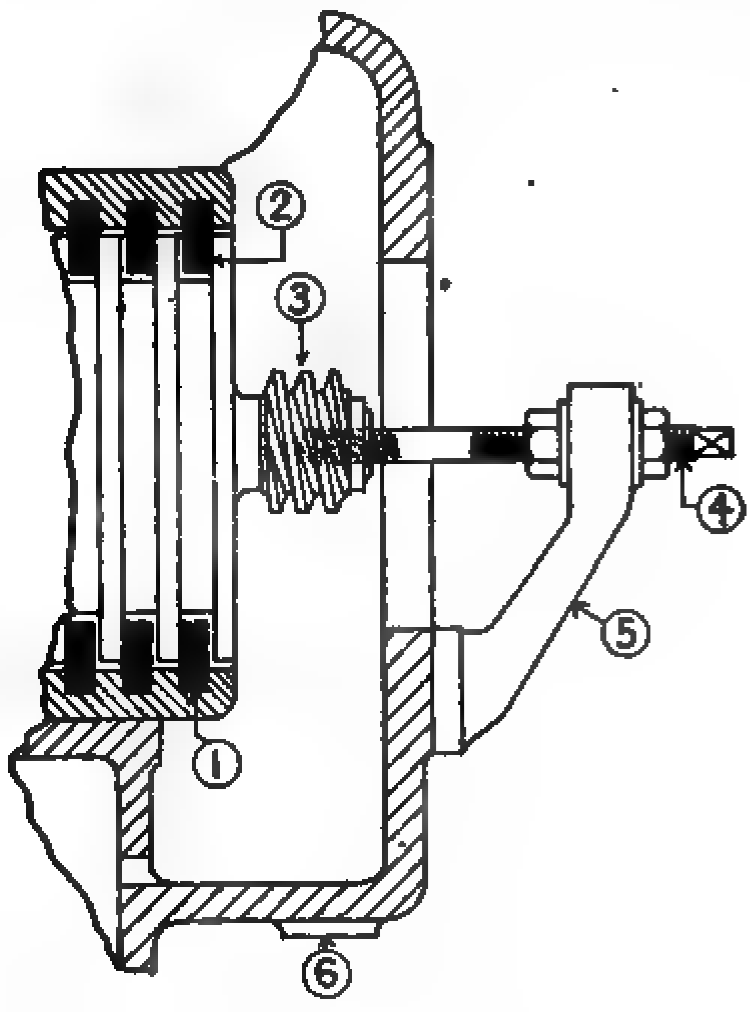
turbines, and any solid matter contained in the steam is trapped and collected in the strainer casing; their purpose is to prevent any solid matter from passing into the interior of the turbine. The strainer cartridge, as it is called, consists of a framework covered with brass perforated plates or of brass grids (see sketch).

Steam Strainer.

1. Steam Inlet. 2. Brass Wire Mantle or Cartridge. 3. Steam Outlet to Turbine.

Calibration of Turbine Shafts.—To calculate the shaft horse-power by torsion meter it is requisite to obtain a constant for the torque of the shaft as this varies with different makes and size of shafting. To attain this result the shafting requires to be calibrated, and the usual method of doing this is as follows:—The shafting between the turbine and tail shaft is laid along the shop floor and bolted together by the coupling bolts belonging to the shaft. This length of shafting is levelled up, and is laid on wood blocks faced with sheet iron. The forward end of the shaft is rigidly bolted to a lever or arm, and on the end of same large weights are laid. At the other or after end of the shaft another lever is bolted, care being taken to see that in both cases there is no play in the holes. This lever is adjusted so that it travels an equal distance above and below the centre of the shaft when the weights are applied. At equal distance along the shaft light clamps are fixed, which have extended arms, on the end of which is fixed a piece of planed wood. The distance from centre of shaft to the wood varies according to size of shaft. These arms are also levelled, and in front of them suitable platforms are arranged on which to work a surface gauge. The movable end of the shaft is

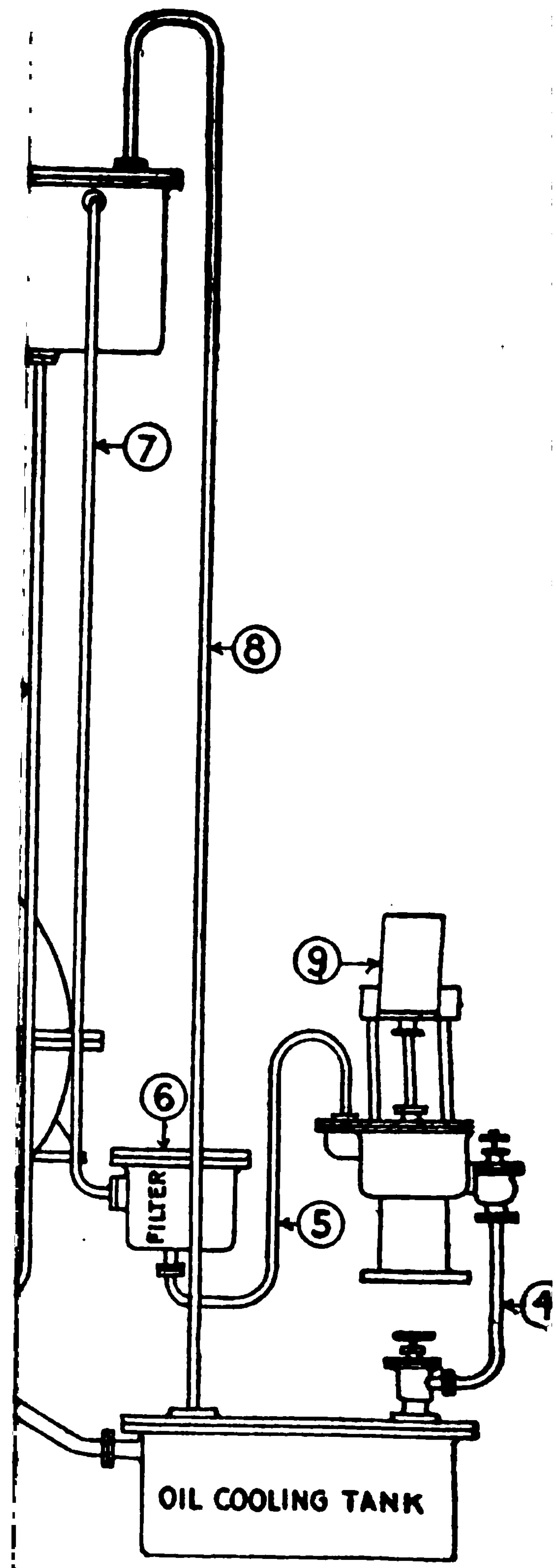
left unweighted, and lines are drawn across the wood on the end of the arms. Weights are now put on the lever at aft end of shaft, these weights being applied at a given distance from centre of shaft, and when sufficient weight has been applied other lines are drawn across the wood on the end of arms. More weight is applied and more lines drawn as the weight is increased. The distances between the marks red and indicate the torque or twist of the shaft due to t applied at the leverage selected. From this operation are obtained which are used in the formula employed in the shaft horse-power by torsion meter readings.



"Screwing-up" Bolt.
d Thrust Rings. 2. Astern Thrust Rings. 3. Worm for Counter Gear. 4. Screwing-up Bolt. 5. Bracket. 6. Oil Drain.

Forced Lubrication in Turbines.

are several systems of forced lubrication. The simplest nt is to have the pump drawing direct from a supply tank, filter, and discharging direct into the bearings, and then ack to supply tank. In another system, the pump draws ıpply tank and discharges through a filter, which consists of ıving inside two brass perforated plates, with a fine gauze ween: the oil passes from the filter to an overhead supply d high up in the engine-room, and on the top of the tank



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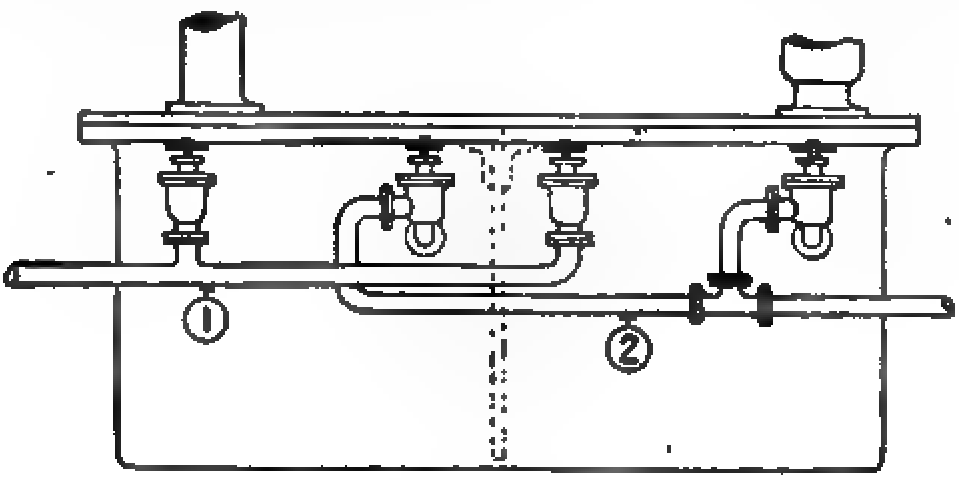
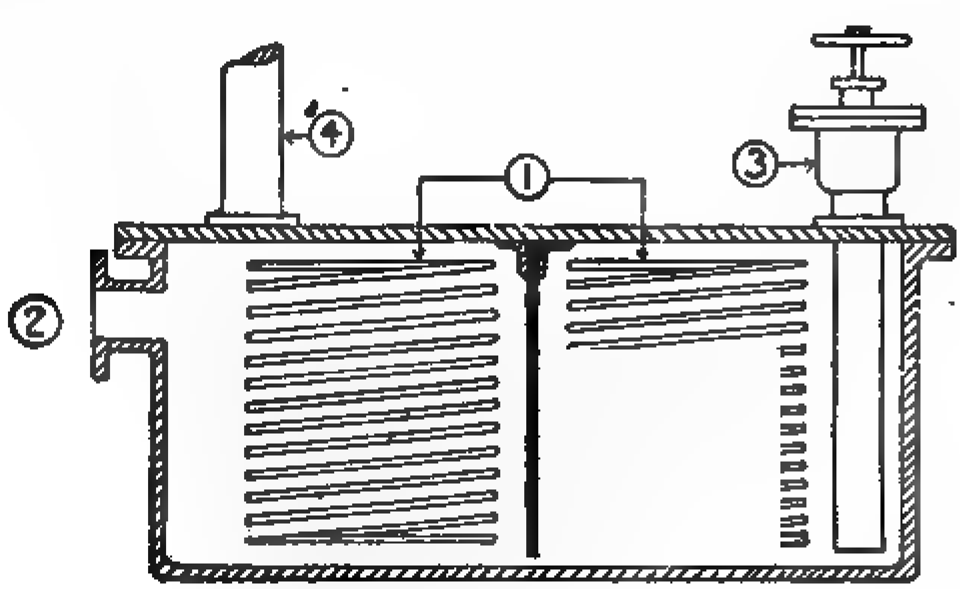
is an overflow pipe, which is led back to the suction tank down below: the oil flows from the overhead tank to the bearings by gravitation, and is then led back by drains to the suction tank below. It is preferable to have the supply tank on top of sufficient size so as to ensure that, in



Oil Cooling Tank.

1. Circulating Water Inlet to Coils. 2. Circulating Water Discharge from Coils.

the event of the pump breaking down, a sufficient supply of oil will be led to turbine bearings, until turbine is stopped or reserve pump started. The suction tank is usually fitted with cooling coils, through which water passes from a connection on the main circulating inlet. This ensures



Oil Cooling Tank (Section).

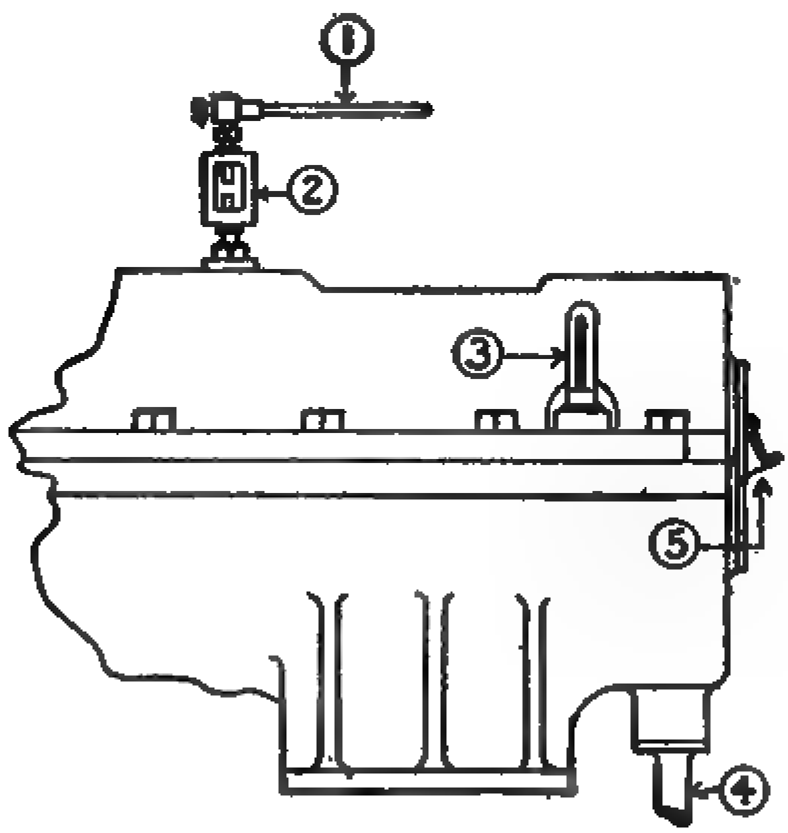
1. Cooling Water Coils.
2. Oil Inlet from Bearings.
3. Suction Valve to Pump.
4. Overflow from Supply Tank Overhead.

sures the oil being kept at as low a temperature as possible, as, owing to the heat imparted to the shaft of the turbine, the oil is soon heated up. On the main bearing covers test cocks are fitted so as to allow of inspection to see that oil is circulating, and hand access doors are arranged at the forward and aft end of the cover to allow of the bush

being felt by hand. Sight drains are sometimes fitted, consisting of a lantern casing having glass inserted in the sides, through which the oil may be observed as it drains back to tank. Gauges are also fitted to the bearings to indicate the pressure on same. This is usually from 4 to 10 lbs. per sq. in. In large bearings thermometers are fitted, showing the temperature of the oil which passes through the bearings.

It is of the utmost importance that the oil service be absolutely
nd uninterrupted, as if not, complete disablement of the
ıay result.

regular maintenance of the oil service is of the highest
ce for the successful running of the turbines, as should



Oil Service to Bearings.

1. Oil Inlet.
2. Sight Feed Glass.
3. Thermometer for Oil Outlet Temperature.
4. Oil Drain.
5. Inspection Door.

ıpply cease, rapid heating up and melting out of the main
would result. This, again, by the subsequent wear-down,
oduce stripping of the vanes of the rotor and drum. As
ntioned, a sight glass, through which the oil passes, is often
a small cock with a hole bored out through the shell so
il spurts out when the cock is turned.

advisable, however, not to use the same oil for too long a
oil repeatedly used has its lubricating properties somewhat
so that fresh supplies of unused oil become a necessity.

ılty is occasionally found in keeping down the temperature
l supply, which, if once heated up, has a decided tendency

H.P. Turbine and one L P. Turbine of S.S. "Tenyo Maru" (Parsons' Turbines.)

(Oriental Steamship Co. of Japan.)

Showing the Screw Adjusting Gear, Governor, Bye-Pass Valves on H.P., Non-Return Valve on L.P., Micrometer Dummy Gauges, and Relief Valves, &c.

H.P. Drum, 76 in. diameter; L.P. Drums, 106 in. diameter; Reverse Drums, 87 in. diameter.

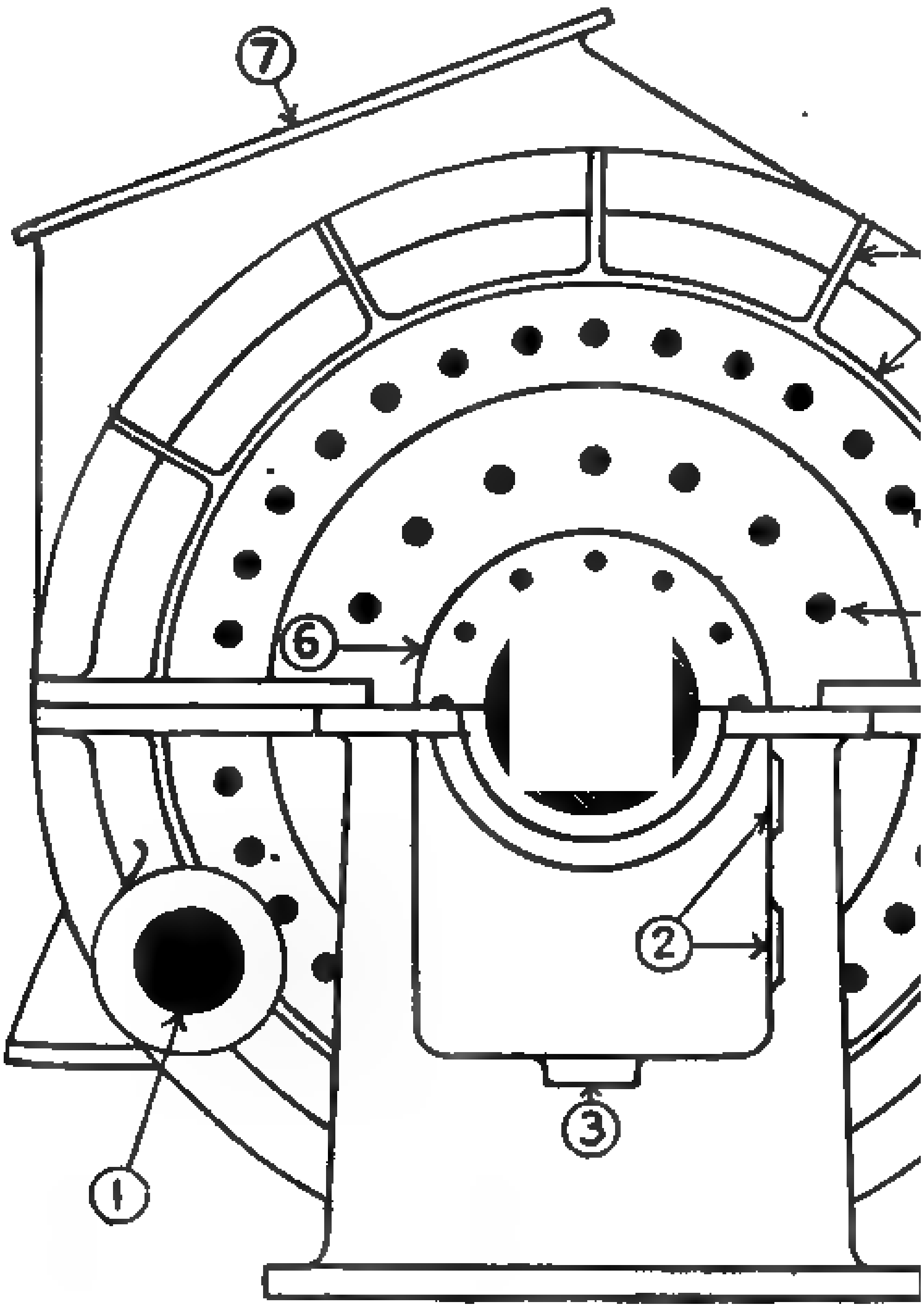
Speed on Measured Course Trials, 20.62 knots.

[*To face page* 136.

to remain very warm, the cooling water having little apparent effect in lowering the temperature.

NOTE.—The specific heat of oil is .36.

Oil is only used in the bearings at each end



End View (Aft) of L.P. Turbine Casing.

1. Astern Steam Inlet.
2. Oil Connections.
3. Oil Drain.
4. Bolts of Astern Casing.
5. Bolts of Astern Dum
6. Gland Case.
7. Exhaust to Condense
8. Stiffening Webs.

and is prevented from entering the turbine casing rings or "deflectors." The condensed steam, theref oily matter than usual, with the resulting advantage less danger of collapsed furnaces and reduced pitting

SECTION III.

DATA FROM ACTUAL PRACTICE.

No. 1. DATA.

Steam flow = 4000 lbs. per minute.
Mean revolutions = 340.
H.P. rotor diameter = 63 in.
L.P. rotor diameter = 90 in.
Constant for blade rows = 1500000.
Constant for H.P. initial steam velocity = 3000.
Initial pressure = 172 lbs. gauge.

Then, V_t (H.P.) $= \dfrac{63 \times 3.1416 \times 340}{60 \times 12} = 93.4$, say 94 ft. per sec.

No. of blade rows $= \dfrac{1500000}{94^2} = 169$ (total).

?. turbine blade rows $= 169 \div 3 = 56$ rows.

efore $56 \div 4$ expansions $= 14$ blade rows for each H.P. expansion.

Again, V_t (L.P.) $= \dfrac{90 \times 3.1416 \times 340}{60 \times 12} = 133.5$ ft. per sec.

No. of blade rows $= \dfrac{1500000}{133.5^2} = 84$.

L.P. turbine blade rows $= \dfrac{84 \times 2}{3} = 56$.

refore $56 \div 8$ expansions $= 7$ blade rows for each L.P. expansion.

P. initial steam velocity $= \dfrac{3000}{\sqrt{169}} = 230$ ft. per sec.

steam volume $= 2.42$ cub. ft. per lb. (187 lbs. absolute pressure).

Cubic feet flow per sec. $= \dfrac{4000 \times 2.42}{60} = 161.3$.

lear area through blades $= 161.3 \div 230 = .701$ sq. ft.

Annulus area $= .701 \times 3 = 2.103$ sq. ft.

" " $= 2.103 \times 144 = 302.832$ sq. in.

1, diameter over blades $= \sqrt{\dfrac{63^2 \times .2854 + 302.832}{.7854}} = 65.9$, say 66 in.

heights (1st expansion) $= \dfrac{66 - 63}{2} = 1\frac{1}{2}$ in.

Assuming a blade height ratio of 1.42 the blade heights of the other expansions may be found as follows :—

2nd expansion, $1.5 \times 1.42 = 2.13$, say $2\frac{1}{4}$ in. blade height.
3rd „ $2.25 \times 1.42 = 3.19$, „ $3\frac{1}{4}$ in. „ „
4th „ $3.25 \times 1.42 = 4.61$, „ $4\frac{1}{2}$ in. „ „

L.P. Blades.

Ratio of rotor diameters $= 90 \div 63 = 1.43$ (nearly).

$$\text{Blade heights (1st expansion)} = \frac{4.5 \times 1.42}{1.43 \times 1.43 \times 2} = 1\tfrac{1}{2} \text{ in.}$$

The blade lengths at the 2nd, 3rd, and 4th expansions will be the same as for the H.P., that is $2\frac{1}{4}$ in., $3\frac{1}{4}$ in., and $4\frac{1}{2}$ in.

5th expansion $= 4.5 \times 1.42 = 6.39$, say $6\frac{1}{4}$ in.
6th „ $= 6.39 \times 1.42 = 9.07$, „ 9 in.

The blades of the 7th and 8th expansions are also 9 in. long, but these blades are of the "wing" type which allow of a much larger exit area than the normal blades of the other expansions.

Steam Pressures and Volumes.

Initial Pressures, &c., at H.P. Expansions.

No.	Gauge Pressure.	Absolute Pressure.	Specific Volume in Cub. Ft.	Actual Volume in Cub. Ft.	Dryness.
1	172 lbs.	187 lbs.	2.420	2.400	1
2	112 „	127 „	3.462	3.402	.982
3	68 „	83 „	5.167	4.968	.961
4	43 „	58 „	7.238	6.860	.947
Terminal Pressure. 4	28 „	43 „	9.590	9.00	.938

Initial Pressures, &c., at L.P. Expansions.

1	25 lbs.	40 lbs.	10.267	9.595	.934
2	12 „	27 „	14.863	13.698	.921
3	3 „	18 „	21.766	19.742	.907
4	6 in. vac.	12 „	31.879	28.400	.891
5	14 „	8 „	46.680	40.940	.877
6	18 „	6 „	61.201	52.987	.865
7	22 „	4 „	89.632	76.608	.854
8	25 „	2.5 „	139.488	117.711	.843
Terminal Pressure (Condenser). 8	27 „	1.5 „	225.580	186.045	.824

NOTE.—The "actual steam volumes" are calculated from the "error diagram" of Mr E. M. Speakman, and on referring to the diagram it will be noticed that the blade height curve approximates to that of the actual steam volumes at the last L.P. turbine expansions.

o of Volumes.—At the various expansions the ratio of steam volumes is nearly the same as the ratio of blade which the following will make clear.

Turbine.—At the 1st and 2nd expansions the specific steam are 2.400 cub. ft. and 3.46 cub. ft.

Then, $3.46 \div 2.42 = 1.42$ ratio.

Turbines.—At the 6th and 7th expansions the specific steam are 61.20 cub. ft. and 89.632 cub. ft.

Then, $89.632 \div 61.20 = 1.46$ ratio.

nilar ratio maintains between the respective actual volumes.

ıber of Steam Expansions.—The total number of times the xpands throughout the turbines *by pressures* is found by the H.P. initial absolute pressure by the L.P. terminal pressure.

'ore, Number of expansions by pressures $= 187 \div 1.5 = 124$ times.

nd the number of steam expansions by volumes divide the ninal actual volume by the H.P. initial volume.

erefore, Expansions by volumes $= 186.045 \div 2.4 = 77.5$ times.

o of V_t to V_s.—The ratio of blade speed to steam speed at ansion may be determined as follows :—

First H.P. Expansion.

Blade speed, $V_t = \dfrac{(63 + 1.5) \times 3.1416 \times 340}{60 \times 12} = 95.6$ ft. per sec.

Steam speed, $V_s = 230$ ft. per sec.

Ratio $= 95.6 \div 230 = .415$.

Fourth H.P. Expansion.

$$V_t = \frac{(63 + 4.5) \times 3.1416 \times 340}{60 \times 12} = 100 \text{ ft. per sec.}$$

Steam flow per sec. $= \dfrac{4000 \times 6.86}{60} = 457.3$ cub. ft.

r area between blades $= \dfrac{(63 + 4.5) \times 3.1416 \times 4.5}{3 \times 144} = 2.2$ sq. ft.

Then, $V_s = 457.3 \div 2.2 = 208$ ft. per sec.

Ratio, $V_t, V_s = 100 \div 208 = .48$.

First L.P. Expansion.

Blade speed, $V_t = \frac{(90 + 1.5) \times 3.1416 \times 340}{12 \times 60} = 135.7$, say 136 ft. per sec.

Steam flow per sec. $= \frac{2000 \times 9.595}{60} = 319.8$, say 320 cub. ft.

Clear area between blades $= \frac{91.5 \times 3.1416 \times 1.5}{3 \times 144} = .99$, say 1 sq. ft.

Then, $V_s = 320 \div 1 = 320$.
Ratio, $V_b\ V_s = 136 \div 320 = .42$.

Fifth L.P. Expansion.

Blade speed, $V_t = \frac{(90 + 6.5) \times 3.1416 \times 340}{12 \times 60} = 143$.

Steam flow per sec. $= \frac{2000 \times 40.94}{60} = 1364.6$ cub. ft.

Clear area between blades $= \frac{(90 + 6.5) \times 3.1416 \times 6.5}{3 \times 144} = 4.56$ sq. ft.

Then, $V_s = 1364.6 \div 4.56 = 299$ ft. per sec.
Ratio, $V_b\ V_s = 143 \div 299 = .48$ nearly.

NOTE.—In the foregoing calculations the actual volume of the steam has been taken at each expansion, and normal blades assumed, giving about one-third annulus area opening for steam flow.

Water Condensed in Turbines.—The water condensed by the nature of the expansion and drained off from the L.P. turbines per hour by the "wet" air pumps may be approximated as follows:—

Dryness at last L.P. expansion $= .824$.
Water condensed in turbines per min. $= 4000 \times (1 - .824) = 704$ lbs.
Water condensed per hour $= 704 \times 60 = 42240$.

Heat Drop and Horse-Power.

H.P. Initial 187 lbs. absolute $= 376°$ Temp. Fahr. Latent heat $= 847.6$.
$376 + 461 = 837$ absolute Temp. Dryness $= 1$.
L.P. Terminal 1.5 lbs. absolute $= 116°$ Temp. Fahr. Latent heat $= 1033.2$.
$116 + 461 = 577$ absolute Temp. Dryness $= .824$
Then, Heat drop $= 847.6 \times 1 - 1033.2 \times .824 + 837 - 577 = 256.25$ B.T.U. per lb.
The mean blade efficiency assumed as 73 per cent. or .73.

Therefore, Horse-Power $= \frac{4000 \times 256.25 \times 778 \times .73}{33000} = 17640$ Horse-Power.

The equivalent I.H.P. $= 17653 \div .9 = 19600$.

NOTE.—The ratio of brake or shaft horse-power to I.H.P. is about .9 to 1.

Pressure Drops.—The following are the pressure drops which occur in each expansion of the foregoing case: assuming that the

terminal pressure of one expansion is the initial pressure of the next expansion, which is approximately correct:—

H.P. Expansions.	Absolute Initial Pressure.	Absolute Terminal Pressure.	Difference, or "Drop."
1	187 lbs.	127 lbs.	60 lbs.
2	127 "	83 "	44 "
3	83 "	58 "	25 "
4	58 "	40 "	18 "
...	...	...	147 lbs. total drop.

L.P. Expansions.	Absolute Initial Pressure.	Absolute Terminal Pressure.	Difference, or "Drop."
1	40 lbs.	27 lbs.	13 lbs.
2	27 "	18 "	9 "
3	18 "	12 "	6 "
4	12 "	8 "	4 "
5	8 "	6 "	2 "
6	6 "	4 "	2 "
7	4 "	2.5 "	1.5 "
8	2.5 "	1.5 "	1 "
...	...	...	38.5 lbs. total drop.

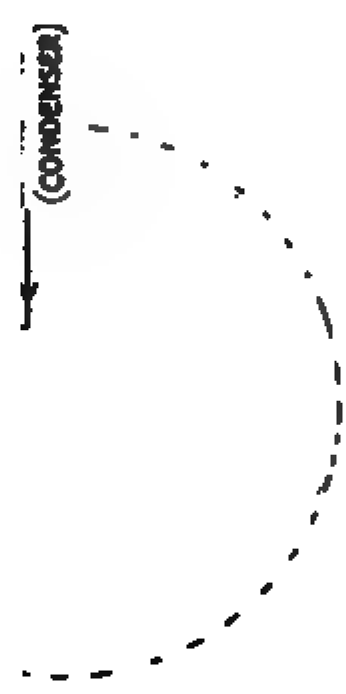
Mean pressure drop per blade row in H.P. turbine = 147 ÷ 56 rows = 2.62 lbs.
" " " " each L.P. " = 38.5 ÷ 56 " = .68 "

No. 2. Description and Trials of U.S.S. "Chester."

The writer is indebted to the President and Council of the American Society of Naval Engineers for permission to reproduce the following descriptions and trial data of the "Chester," from the pages of the *Journal* issued quarterly by the Society.

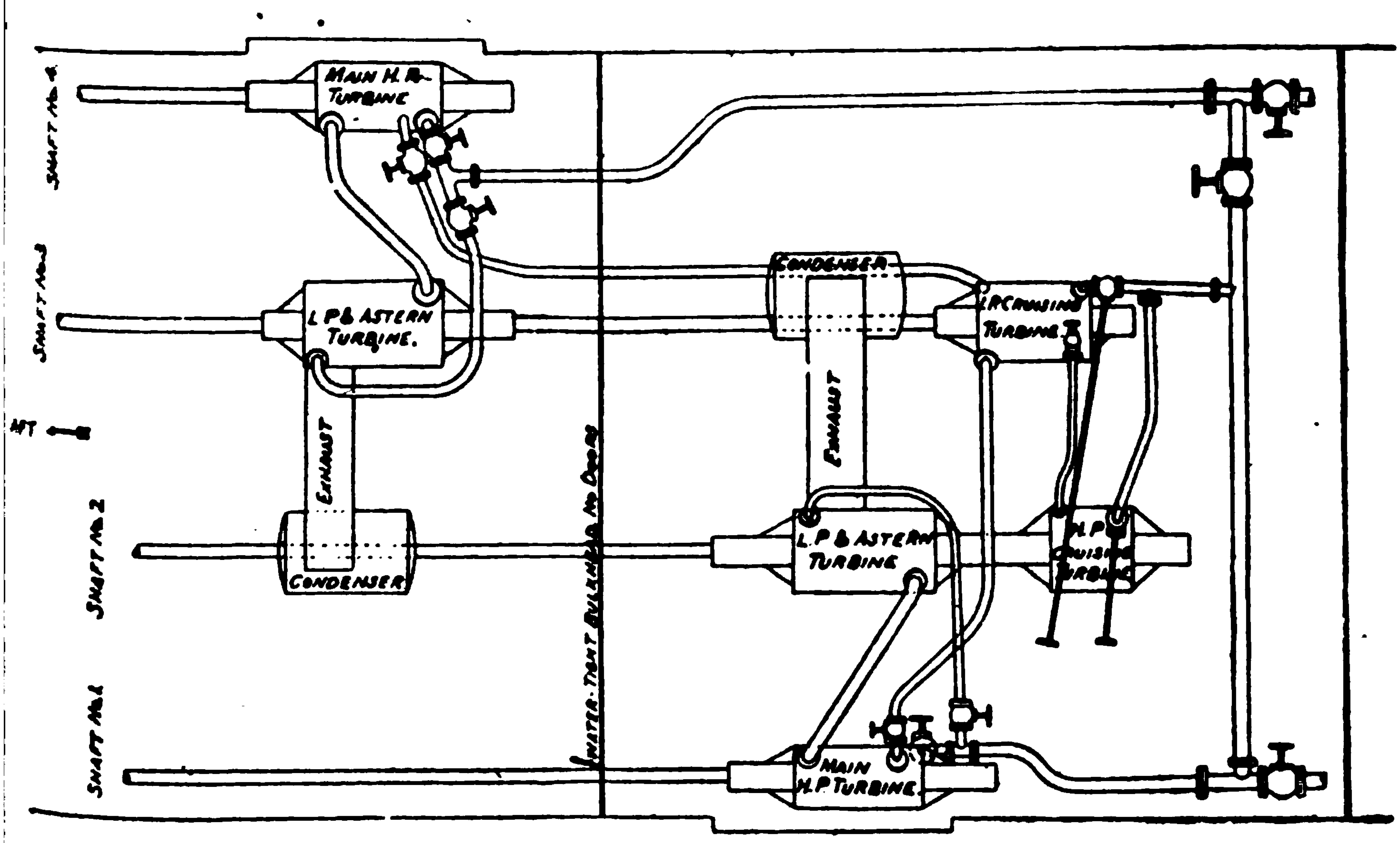
Displacement - - - - - - - -	3,750 tons.
I.H.P. (designed) at 24 knots - - - - -	16,000.
Length (b. p.) - - - - - - - -	420 ft. 0 in.
" (over all) - - - - - - - -	423 ft. 2 in.
Beam - - - - - - - - - -	47 ft. $1\frac{1}{8}$ in.
Draught - - - - - - - - -	16 ft. $8\frac{9}{16}$ in.
Ratio of length to beam - - - - - -	8.9
Number of screws - - - - - - -	4
Wing shafts incline up and forward - - - -	2.46 deg.
Centre " " " " - - - -	1.44 deg.
Shafts diverge from centre line of ship - - - -	.72 degree.

NOTE.—The horse-power developed at the limit speed was much in excess of the 16,000 horse-power designed.



page 142.

The propelling machinery consists of six Parsons' marine turbines driving four independent shafts, each shaft being fitted with one propeller; two cruising turbines are fitted to give fan economy at low speeds and powers. Exclusive of the cruising turbines there are two main high-pressure turbines exhausting into two low-pressure turbines, and in each of the latter there is incorporated a reversing turbine. To obtain economy at low and moderate speeds the six turbines may be used in three combinations.



Turbine Arrangement.
U.S. Cruiser Scout "Chester."

(1) **Low Speeds** (up to 18 knots).—The steam passes through all six ahead turbines, both the H.P. cruising and the I.P. cruising being connected up with the four main turbines. Steam admitted to the H.P. cruising turbine exhausts into the I.P. cruising turbine, and from the latter exhausts through separate pipes to each of the main H.P. turbines. From these latter it exhausts into the L.P. turbines and then into the main condensers.

(2) **Moderate Speeds** (up to 23 knots).—The steam passes through five ahead turbines, steam being admitted to the I.P. cruising turbine and passing thence to the two main high-pressure turbines, and from each of them to the connected low-pressure turbine. The high-pressure cruising turbine revolves idly in a vacuum.

(3) **Highest Speeds**.—Only the four main turbines are used, steam being admitted to each main H.P. turbine. Both cruising turbines revolve idly in a vacuum.

Reduction of power in each of the arrangements is obtained by throttling. Increased power may be obtained by admitting live steam to the I.P. cruising turbine in the *first* arrangement and to the main H.P. turbine in the *second* arrangement. Bye-pass valves are fitted from the 1st to the 2nd expansions in both main H.P., and auxiliary exhaust steam may be admitted to the 2nd expansions in main H.P. turbines and to both L.P. receiver pipes.

The turbines were designed for an initial working pressure of 250 lbs. per sq. in., to drive the shafts at 502 revolutions per minute for the contract speed of 24 knots.

Description of Propelling Machinery.

The propelling turbines are six in number, an H.P. cruising, I.P. cruising, two main high pressure and two low pressure, with reversing turbines incorporated into the exhaust ends of each of the latter. The reversing turbines revolve idly in the exhaust casing of the L.P. turbine when the engines are running ahead. The turbines are located in two compartments separated by an athwartship, watertight bulkhead. The inboard shafts turn outboard and the outboard shafts turn inboard.

The turbine cylinders are parted horizontally in the plane of the shaft, and the two halves strongly bolted together. The lower half is cast with extensions forward and aft, box shaped, for retaining the journal and thrust bearings. The cylinders are supported by feet, certain of which are securely bolted to the foundations and others have slotted holes for expansion. The feet are at a point directly under the journal bearings. The cylinders contain supports for the spindle and thrust-bearing brasses with oil pockets, also the lower halves of spindle glands, and have pockets under these glands for steam leak-offs. They have internal facings at their ahead steam ends for bolting on the dummies. Cylinder blading is caulked into grooves on the inside of the cylinder castings; these portions of the cylinders have been accurately bored and finished in a horizontal position. Hydrostatic tests were first applied to the cylinders, and subsequently a "baking" under moderate steam pressure for forty-eight hours to relieve internal stresses in the castings. The following table shows arrangement and dimensions of blading.

The rotors were statically balanced on two truly-levelled rails and the balance adjusted by removing metal, as necessary, from "chipping strips" left on the arms of the rotors for the purpose. The dynamic balancing was done by revolving the rotors under steam. At the same time dummy ring faces were worn down, and bearings adjusted.

Blading.

Expansions.	Rows.	Heights, Inches.	Pitch, Inches.	Clearances, Inches. (Cold.)
H.P. Cruising Turbine.				
1st	12	$\frac{3}{8}$	$\frac{7}{8}$	.03
2nd	12	$\frac{1}{2}$	$\frac{7}{8}$	.035
3rd	12	$\frac{5}{8}$	1	.04
I.P. Cruising Turbine.				
1st	15	$1\frac{1}{8}$	$1\frac{1}{4}$	.04
2nd	15	$1\frac{7}{16}$	$1\frac{1}{4}$	.04
3rd	15	$1\frac{3}{4}$	$1\frac{5}{16}$	.045
High-Pressure Turbines.				
1st	12	$\frac{7}{8}$	$1\frac{1}{4}$	.03
2nd	12	$1\frac{1}{4}$	$1\frac{1}{4}$	.035
3rd	12	$1\frac{3}{4}$	$1\frac{5}{16}$	.04
4th	12	$2\frac{1}{2}$	$1\frac{3}{8}$	.045
5th	12	$3\frac{1}{2}$	$1\frac{7}{16}$	.05
6th	10	5	$1\frac{13}{16}$	.05
Low-Pressure Turbines.				
1st	5	$2\frac{1}{2}$	$1\frac{5}{16}$	.055
2nd	5	$3\frac{1}{2}$	$1\frac{3}{8}$	.06
3rd	4	5	$1\frac{13}{16}$	.07
4th	3	7	$1\frac{3}{4}$	.08
5th	3	7	$1\frac{15}{16}$	.08
6th	4	7	$2\frac{5}{16}$	.08
7th	4	7	$2\frac{5}{16}$	.08
Astern Turbines.				
1st	6	$\frac{1}{2}$	$1\frac{1}{8}$	.045
2nd	6	1	$1\frac{3}{8}$	.05
3rd	6	2	$1\frac{3}{8}$	.06
4th	6	2	$1\frac{1}{2}$	.06
5th	6	2	$1\frac{1}{2}$	.06

Turbine Cylinder Data.

Cylinder.	Diam. of Rotor Drum.	Length of Rotor Drum.	Diameter of Cylinder for each Expansion.						
			1st.	2nd.	3rd.	4th.	5th.	6th.	7th.
	In.	In.	In.	In.	In.	In.	In.	In.	In.
H.P. cruising	60	36	$60\frac{3}{4}$	61	$61\frac{1}{4}$	...	...	...	...
I.P. cruising	49	60.5	51	$51\frac{7}{8}$	52.5	...	...	...	...
Main H.P. port	42	103.5	$43\frac{3}{4}$	44.5	45.5	47	49	52	...
" " starboard	42	103.5	$43\frac{3}{4}$	44.5	45.5	47	49	52	...
" L.P. port	65	$53\frac{3}{8}$	70	72	75	79	79	79	79
" " starboard	65	$53\frac{3}{8}$	70	72	75	79	79	79	79
Astern, port	50	43	51	52	54	54	54	...	...
" starboard	50	43	51	52	54	54	54	...	...

NOTE.—The diameter of cylinder at each expansion is obtained by adding twice the blade height to the rotor drum diameter.

"The blades were installed in accordance with the Parsons system. They were cut to length, saw-cuts made where required for binding, caulking grooves stamped into the base of the blades, all done by the same machine, and the blades then caulked into place. Caulking strips having curves corresponding to the blades were cut to length and placed alternately at the base of each blade. The blades and caulking strips were alternately tightened into the groove by means of a caulking tool, after which the binding wire was inserted and the lacing put on around binding wire and blades with silver solder. Blades were turned up in the vertical line before the insertion of the binding wire, and were finally filed up to remove burrs.

"The blades are of a composition consisting of 72 per cent. copper and 28 per cent. zinc, the caulking strips $63\frac{1}{2}$ per cent. copper and $36\frac{1}{2}$ per cent. zinc, the binding strip 72 per cent. copper and 28 per cent. zinc, and the lacing wire pure copper. Blades having binding strips are soldered to the latter with a silver solder. Steam supply and exhaust piping is as shown on page 143. In the exhaust pipe from H.P. cruising turbine to I.P. cruising turbine, and in the two exhaust pipes from the I.P. cruising to the main H.P. turbines, spring-loaded, self-closing valves are installed to isolate the cruising turbines while running at higher rates of speed and while manoeuvring. A $3\frac{1}{2}$-in. spring-relief valve for the H.P. cruising turbine is installed in its exhaust pipe, set at 125 lbs., and one for the I.P. cruising turbine is installed on after end of its upper casing, set at 160 lbs. A governor, arranged to pull out a pawl and close a butterfly valve in the main steam pipe, is fitted on each shaft, the object being merely to shut down automatically in the event of accident. An unusual feature in piping is the low-pressure exhaust pipe, which is an elbow pipe rectangular in cross section, 3 ft. 9 in. by 4 ft. 6 in. and of $\frac{3}{8}$-in. steel plate. The dummies in the steam ends of turbines for preventing steam

from leaking from steam belts direct to the inside of drums consist of a series of rectangular grooves in the spindle wheels against the side of each of which a brass strip lies closely without touching. These brass strips are caulked into an annular iron casting made in halves and bolted to the inside of the cylinder. A micrometer for measuring the opening of dummies between brass strips and running steel faces is fitted on each cylinder. Steam glands are fitted around turbine spindles at cylinder ends, packed with labyrinth stuffing boxes. These glands are to obviate, in the case of high-pressure turbines, the leakage of steam from the casings, and in the case of the low-pressure turbines, the leakage of air from the atmosphere into the casing and thence to the condenser. There are rows of brass strips let into the shaft and the cast-iron gland sleeve, with thin edges just clearing the opposite member. Snap rings of composition H are fitted at the outer end of each gland box, and just inside of these rings connection is made with an equaliser pipe joining all similar pockets, and a valve is provided from auxiliary exhaust line for maintaining the pressure in this pipe. Strainers of the basket type in accordance with current turbine practice, made with bodies and covers of composition G, and with strainer baskets of sheet brass, are installed, one in the live-steam supply to each turbine and one in the auxiliary exhaust connection to the 2nd expansion of the main H.P. turbines and L.P. turbines. A system of guide rods is installed for lifting turbine casings, one on each of the four corners, placed vertically and graduated in inches, to permit of the even lifting of the covers. Special portable guides are supplied for guiding the spindles when lifting them, to prevent the stripping of blades. Turning gear is fitted at the after end of each turbine for turning it by hand. It consists of a worm wheel on the shaft, meshing with a worm operated by a ratchet wrench. Suitable lifting gear is provided for lifting casings, spindles, &c., consisting of overhead trolleys on frames, with chain falls and slings."

Shafting and Shaft Bearings.—"The shafting is of Class 'A' forgings and is solid. The thrust shaft is on spindle ends. The stern-tube shaft and propeller shaft is in one piece, inboard sections 48 ft. 3 in. long, and outboard lengths 46 ft. $9\frac{11}{16}$ in. long, in both cases $8\frac{1}{2}$ in. diameter. The line shafting is 8 in. in diameter. There are two shaft bearings to each turbine—twelve in all—those for the cruising turbines being 10 in. long, and those for the main turbines 15 in. The bearing shells, of brass, are semicircular, and are lined with white metal. Broad strips of the bearing shell are exposed slightly below the line of the white metal to catch the spindle in case the white metal gives out. The bottom half can be rolled out without lifting the spindle, the top half will lift off. These bearings are lubricated by oil supplied under pressure. There are six thrust-shaft bearings, those for the cruising turbines having eight collars and rings, those for main turbines fifteen collars These bearings are so constructed that the longitudinal

position of the shaft and the dummy clearance can be readily adjusted. The thrust bearings are in two halves. The steam balance is so arranged as to be practically equal to the thrust of the propeller at all speeds. The thrusts are lubricated by oil under pressure. Oil supplied to thrust bearings and journal bearings is collected in a chamber in the box ends of lower cylinder castings, and drains by gravity into a return pipe. There are fifteen line-shaft bearings each 12 in. long, lined with white metal, and with ends provided with oil baffles; provision is made for their lubrication by oil under pressure. Water service to a jacket around oil chambers is provided only in case of line-shaft bearings, the water service pipe discharges into stern-tube stuffing box, and has a branch pipe for draining the latter. Water service by hose can be provided in emergency. Stern-tube bearings for each shaft consist of a bearing in each end of the tube in each case. They are each 51¾ in. in length. Each shaft also has one strut bearing 51¾ in. in length, and fair-water sleeves are fitted on the forward side of the strut at one end, and on the after side of the stern tube at the other. There are sixteen bulkhead stuffing boxes where shafts pass through watertight bulkheads.

"An expansion coupling is fitted between each L.P. turbine and the cruising turbine on same shafting. The forward end of each L.P. turbine shaft is built with twelve teeth, which engage with a slotted collar bolted to the flange coupling on after end of cruising turbine shafting. This permits of disconnecting cruising turbines, as well as of expansion. As but slight increase efficiency is gained by disconnecting, this is not done as a rule, the cruising turbines being run idle in a vacuum when not in use."

Propellers.—There are four solid, true-screw, manganese bronze propellers, each having three blades; diameter, 6 ft.; pitch, 6 ft.; area, projected, 17.02 sq. ft.; helicoidal, 19 sq. ft.; disc area, 28.27 sq. ft.; immersion of upper tip of blade, 69½ in. inboard, 57½ in. outboard. The driving surfaces of all propeller blades were made a true surface by machining. The wing propellers are ahead of the inner propellers, and the former turn inboard and the latter outboard, though it was originally intended that all four turn outward.

Main Condensers.

There are two main condensers, one in each engine-room, located inboard and abreast of each L.P. turbine. They are horizontal and cylindrical, of the surface-condensing type. The shells are made of Class "B" boiler plate, water chests of composition, tubes of composition, and tube sheets and supporting plates of "rolled naval bronze." The forward water chest, being the one for the entrance and exit of circulating water, has a division plate with a 7-in. bye-pass valve on outside of chest. Each water chest has eight 10-in. man-

holes fitted with composition covers, inside of which are placed zinc plates. The following is the data for each:—

Cooling surface -	8,999 sq. ft.
Number of tubes - - - -	5,630
Tube thickness - - - -	18 B.W.G.
Outer diameter of tubes - - -	$\frac{5}{8}$ in.
Length of tubes - - - -	10 ft. $0\frac{1}{8}$ in.

Augmenter Condenser.

An arrangement called a "vacuum augmenter," devised by the Parsons Turbine Company, is installed for the purpose of increasing the vacuum above that obtained by the air pump.

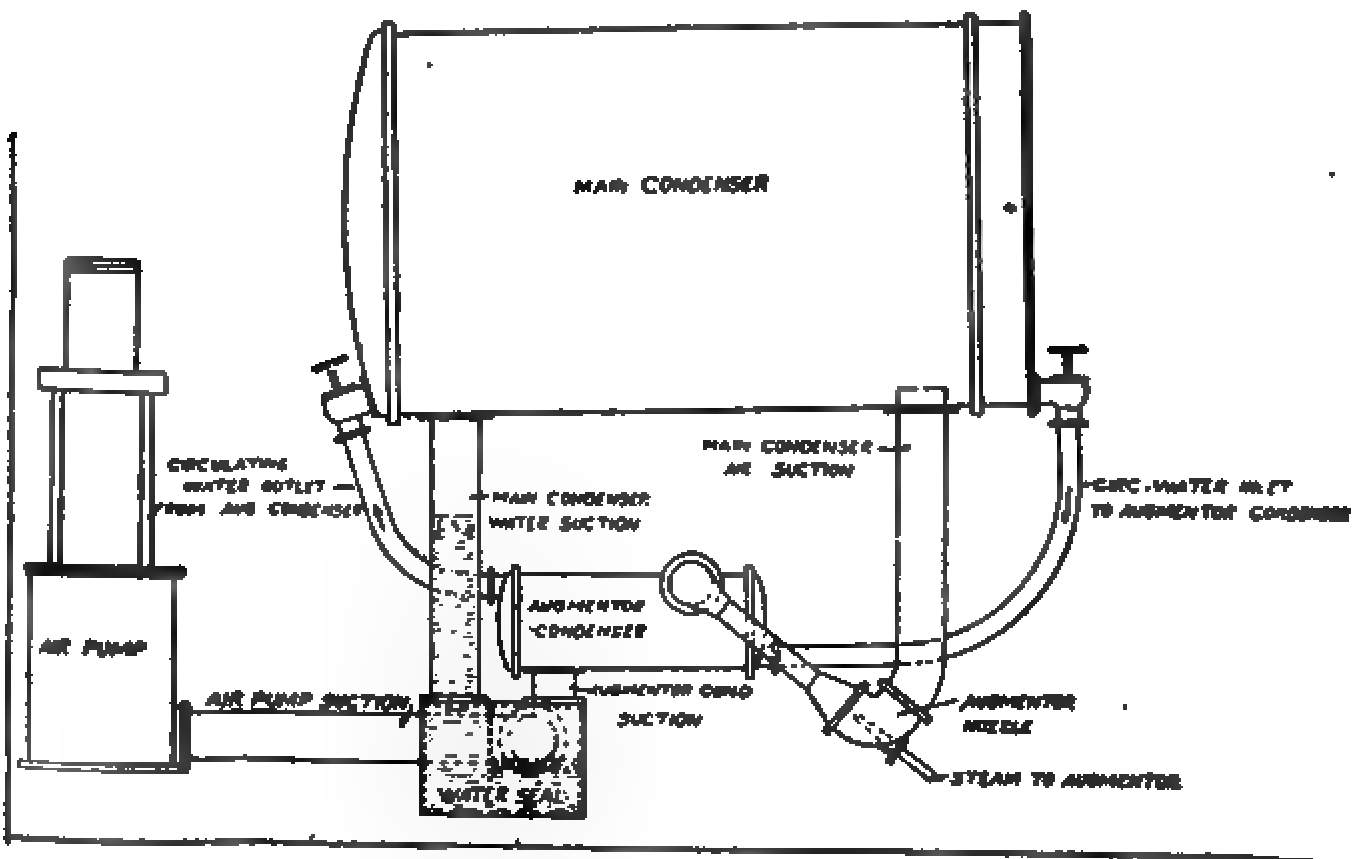


FIGURE 3

Parsons' Vacuum Augmenter.

The augmenter consists of a steam syphon drawing air from the condenser and discharging it to the air pump suction at a pressure from 1 to $1\frac{5}{16}$ in. of mercury higher than the condenser pressure. The discharge from the syphon passes through a small condenser in order to condense the steam of the syphon jet. The air pump has a direct water suction from the condenser through a pipe having a water seal and holding a head of water equal to the difference in pressure produced by the augmenter jet. The augmenter was in use at all times during the trials, and produced an increase of vacuum of from $1\frac{1}{4}$ in.

Lubrication.

The lubrication of all main journal bearings, thrust bearings, and line-shaft bearings is by oil supplied under about 10 lbs. pressure by steam-driven oil pumps. The oil passes through a cooler on its way to the bearings. Tanks of 150 gallons capacity are located in the lower parts of engine-rooms, one in each, from which the oil pumps draw. These tanks take the return of the oil from all bearings supplied with forced lubrication, by gravity. The discharge of the oil from turbine bearings is through glasses so that the flow can be observed, and thermometers are fitted to these discharges. The oil pipes are all of copper, and have no valves fitted either to or from the bearings. Two pumps, Blake, vertical, simplex-piston type, 10×9×12, are supplied, one in each engine-room. Either or both may be used. An oil cooler of the surface-condenser type, with oil passing through the tubes and the water around them, is provided in each engine-room. The oil passes four times the length of the cooler by means of bridges in the oil heads. The circulating water is taken from either the engine-room fire and bilge pumps or main circulating pumps, and discharges through outboard delivery pipes. Additional oil may be supplied to the forced lubrication system by an overhead gravitation system from the tanks containing reserve oil. The outlets for oil in the casings are at a high level, so that there is always in the casing wells a large gathering of oil which is at about the same temperature as the shaft.

Main Air Pumps.

Two independent Blake, vertical, twin-beam air pumps, 14×35×21, are installed, one for each engine-room. The suction openings are 12 in. in diameter, and the discharge 11 in. The air-pump suctions take from the lower end of the vacuum-augmenter condensers and water seals (see Vacuum-Augmenter Condenser and Fig. 3).

Main Circulating Pumps.

For each main condenser there is one main circulating pump, centrifugal, with vertical compound engine of B.I.W. design, steam cylinders, 10 in. and 16 in. in diameter; stroke, 10 in.; 42-in. diameter runner. Each pump is of sufficient power to discharge 11,000 gallons of water per minute at about 360 revolutions. Each pump is fitted with pipes and valves to draw from the sea and the main drain, and to deliver into the condenser or overboard through valves in the condenser water chest. Circulating water for oil coolers may be supplied from these pumps.

Boilers

There are twelve Normand water-tube boilers of the "intermediate" type in the vessel in three watertight compartments, four to each compartment. The following is the boiler data :—

Designed working pressure	250 lbs.
Test pressure	400 lbs.
Ratio of grade surface to heating surface	1 to 46
Length of grates	7 ft. 2⅛ in.
Width of grates	8 ft. ⅞ in.
Per cent. of air space in grates	49
External height	12 ft. 5⅜ in.
" length	11 ft. 6⅝ in.
" width	11 ft. 4 in.
Number of furnaces	1
Grate surface	58 sq. ft.
Heating surface	2,670 sq. ft.
Total grate surface	696 sq. ft.
" heating surface	32,040 sq. ft.
Weight of boiler, with water	19,675 lbs.
Boiler tubes	cold-drawn seamless steel.
Number of tubes each boiler	986, No. 10 B.W.G., O.D., 1⅜ in. lengths vary for each row.
Mean height of smoke pipes above grates	63.25 ft.
Diameter of smoke pipes	72 in.

Performance.

Four Hours' Official Trial.

Steam Pressure.

	Starboard.	Port.
Mean steam pressure at engines, pounds	242.4	246.0
In main H.P. turbine (per gauge)	237.0	240.0
In 2nd I.P. cruising turbine (absolute)	237.0	240.0
In L.P. turbine (absolute)	30.2	29.9
Vacuum in condensers, inches of mercury, mean	28.9	28.3
Average revolutions per minute	614	
Speed of ship, in knots	26.522	
Slip of propeller, in per cent. of its own speed, based on mean pitch. Shafts are numbered from starboard to port.	(1) 25.71 (2) 25.99 (3) (4)	
Air pressure in fire-rooms, in inches of water, mean		

Coal.

Kind and quality used on trial - - - - Pocahontas, |
Pounds, per hour, main and auxiliary engines, during trial -
Pounds of coal per square foot of grate surface, per hour -

Twenty-four Hours' Official Trial at 22.5 Knots.

Steam Pressure.

	Starboard.	Port.
:am pressure at engines, pounds	230.0	
ruising turbine (gauge)	...	192.0
H.P. turbine (gauge)	96.0	93.0
urbine (absolute)	15.0	13.8
in inches	28.3	29.5
revolutions per minute	473	
ship, in knots	22.782	
›ropeller, in per cent. of its own speed, based on .n pitch. Shafts are numbered from starboard to port.	(1) 15.66 (2) 16.94 (3) 26.08 (4) 15.15	
:ure in fire-rooms, in inches of water, mean	0.7	

Coal.

d quality used on trial	Pocahontas, run of mind.
›er hour, main and auxiliary engines, during trial	18,063
:r ton of coal	2.82
›f coal per square foot of grate surface, per hour	25.95

Twenty-four Hours' Official Trial at 12 Knots.

Steam Pressures.

	Starboard.	Port.
:am pressure in engine-room	148.0	
cruising turbine (gauge)	77.3	
ruising turbine (gauge)	...	34.6
H.P. turbine (gauge)	8.0	6.8
:urbine (absolute)	3.3	3.4
in inches	28.4	28.6
revolutions per minute	250	
ship, in knots	12.2	
ropeller, in per cent. of its own speed, based on pitch. Shafts are numbered from starboard to port.	(1) 12.21 (2) 24.66 (3) 19.59 (4) 12.08	

Coal.

d quality used on trial	Pocahontas, run of mine.
›er hour, main and auxiliary engines, during trial	4,091
:r ton of coal	6.68

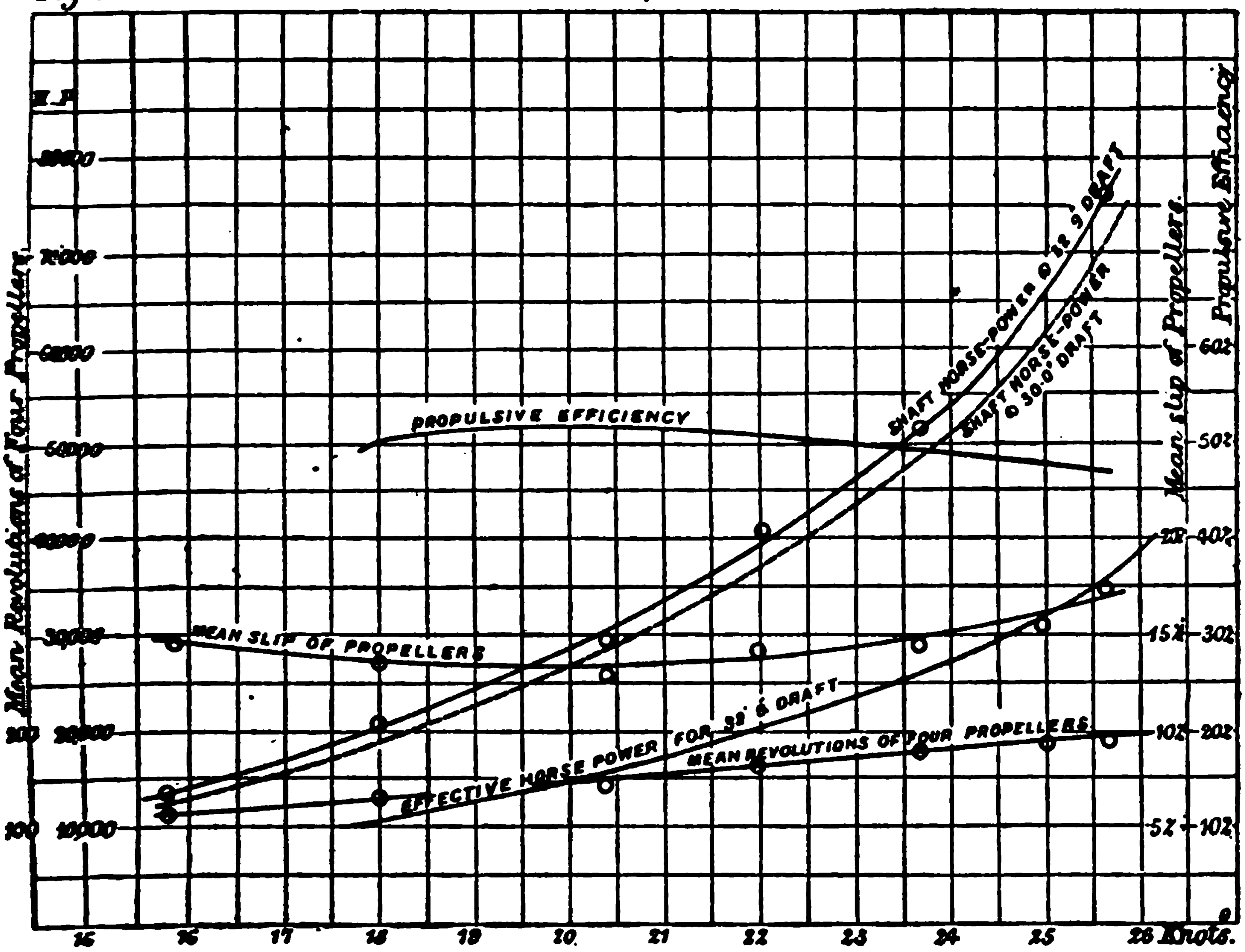
Deduced Data.

›f coal per square foot of grate surface per hour	17.63

t.—Vibration of turbines and hull were practically *nil* at all speeds, remor only being noticed in the after cabin at top speed, a propeller . Both bow waves and stern waves were remarkably small.

3. The following extracts from the paper read by Thomas Bell, Esq., before the Institution of Naval Architects, 9th April 1908, entitled "Speed Trials and Service Performance of the Cunard Turbine Steamer 'Lusitania,'" may prove of interest :—

Fig 6. PROGRESSIVE TRIAL OFF SKELMORLIE, JULY 27, 1907.



Speed Power Curve, "Lusitania."

ENGINES.

Turbines.	Diameter of Rotor.	Length of Blades.	
		In First Expansion.	In Last Expansion.
	In.	In.	In.
High-pressure -	96	$2\frac{3}{4}$	$12\frac{3}{8}$
Low-pressure -	140	$8\frac{1}{4}$	22
Astern - - -	104	$2\frac{1}{4}$	8

TABLE V.

Actual steam and coal consumption of main and auxiliary engines at various speeds under conditions prevailing on official trials, viz., turbo-generators exhausting to auxiliary condensers, other auxiliaries exhausting to heaters.

Shaft horse-power - - - - -	13,400	20,500	33,000	48,000	68,850
Speed in knots - - - - -	15.77	18.0	21.0	23.0	25.4
Temperature of feed water - -	200°	200°	199°	179°	165°
Total consumption of auxiliaries in lbs. per hour - - - - -	71,000	85,700	76,400	96,700	116,500
Total consumption of turbines in lbs. per hour - - - - - -	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries in lbs. per turbine horse-power hour -	5.3	3.72	2.6	2.01	1.69
Steam consumption of turbines in lbs. per horse-power hour - -	21.23	17.24	14.91	13.92	12.77
Total steam consumption in lbs. per horse-power hour - - -	26.53	20.96	17.51	15.93	14.46
Coal consumption in lbs. per horse-power hour - - - -	2.52	2.01	1.68	1.56	1.43
Estimated coal consumption in tons on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, &c. -	2,980	3,190	3,670	4,520	5,390

TABLE VI.

Estimated steam and coal consumption at various speeds, allowing for the additional auxiliary steam consumption found requisite under actual service conditions for the washing water supply, &c., with a full complement of passengers, weather conditions being as on official trial.

Shaft horse-power - - - - -	13,400	20,500	33,000	48,000	68,850
Speed in knots - - - - -	15.77	18.0	21.0	23.0	25.4
Temperature of feed water - -	200°	200°	200°	200°	200°
Total consumption of auxiliaries in lbs. per hour - - - - -	93,500	100,900	112,700	127,500	149,700
Total consumption of turbines in lbs. per hour - - - - - -	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries in lbs. per turbine horse-power hour -	6.97	4.92	3.41	2.65	2.17
Steam consumption of turbines in lbs. per horse-power hour - -	21.23	17.24	14.91	13.92	12.77
Total steam consumption in lbs. per horse-power hour - - -	28.2	22.16	18.32	16.57	14.94
Coal consumption in lbs. per horse-power hour - - - -	2.76	2.17	1.8	1.62	1.46
Estimated coal consumption in tons on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, &c.	3,270	3,440	3,930	4,700	5,490

TABLE VII.

Abstract of Engine-Room Log for third voyage west: Queenstown to New York.

Date when last dry docked, 22nd July 1907. Mean draft, leaving Queenstown, 33 ft. 7 in.
Mean draft, arriving New York, 30 ft. 10 in.

Date, 1907.	Steam Pressures.			Temperatures.		Vacuum.	Barometer.	Length of Day.		Distance by Observation.	Mean Speed.	Mean Revs.	Mean Slip.	Coal Consumed for Main and Auxiliary Engines.
	Boilers.	H.P. Recrs.	L.P. Recrs.	Hot-well.	Feed Water.									
	Lbs.	Lbs.	Lbs.			In.	In.	Hrs.	Mins.	Naut. Mls.			Per Cent.	Tons.
Noon, 3rd Nov. -	170	140.0	2.3	68°	200°	28	30.4	0	52	21	24.24	182.5	16.5	40
Noon, 4th Nov. -	160.1	142.2	2.2	78°	197°	28	29.7	24	57	606	24.28	182.6	16.4	1,090
Noon, 5th Nov. -	167.3	140.6	2.3	78°	198°	28.2	30	25	2	616	24.6	182.8	15.4	1,090
Noon, 6th Nov. -	168.3	149.4	2.5	70°	196°	28.2	30.1	24	55	618	24.8	183.5	15.1	1,090
Noon, 7th Nov. -	168.3	138.3	2.2	72°	195°	28	29.6	24	52	610	24.52	181.4	15.0	1,090
1.14 A.M., 8th Nov. -	165	132.5	1.5	75°	200°	27.8	29.3	14	2	310	22.09	174	20.2	*576
Means - -	168	139.3	2.2	74.5°	197°	28.1	29.8	Total. 114	40	Total. 2,781	24.25	181.1	15.9	4,976

* This includes all coal used till 10 A.M. on the 8th.

Summary of total coal consumed on voyage:—Liverpool to Queenstown, 408 tons; Queenstown to New York, 4,976 tons; galleys, &c., 18 tons—total coal taken from bunkers, from leaving landing stage, Liverpool, till moored at wharf, New York, 5,402 tons. Passage—Queenstown to Sandy Hook—4 days, 18 hours, 40 minutes.

TABLE VIII.

Date, 1907.	Length of Steaming Day.	Distance Run, Nautical Miles.	Speed, Knots.	Total Distance Steamed.	Total Time.	Mean Average Speed.
	Hrs. Min.				Hrs. Min.	
Noon, 3rd Nov. -	0 52	21	24.24	21	0 52	24.24
Noon, 4th Nov. -	24 57	606	24.28	627	25 49	24.27
Noon, 5th Nov. -	25 2	616	24.6	1,243	50 51	24.44
Noon, 6th Nov. -	24 55	618	24.8	1,861	75 46	24.57
Noon till midnight, 6th Nov. - -	12 30	315	25.2	2,176	88 16	24.65
Noon, 7th Nov. -	12 22	295	23.85	2,471	100 38	24.55
Morning, 8th Nov.	14 2	310	22.09	2,781	114 40	24.25

"Table VI. has been compiled for comparison with Table V., to show the additional consumption of steam for auxiliary purposes under actual working conditions at sea with the ship full of passengers. This shows very clearly the demand which modern improvements make on the steam, and hence coal consumption of a large passenger vessel. An additional line has been added to Tables V. and VI. to show total coal consumption on a voyage of 3,100 nautical miles at the various speeds and under the different conditions.

"With reference to the third voyage west, from 2nd November to 8th November of last year, thanks to the courteous permission of the chairman of the Cunard Company, the leading particulars of the official engine-room log are summarised in Table VII. Regarding the mean draft of the vessel at sea, it may be remarked that, after the second day out, certain of the forward tanks were gradually filled for the purpose of avoiding excessive trim, so that the mean draft on 5th, 6th, and 7th November was approximately 32 ft., or very little more than the mean of the first pair of runs from Corsewall Point to the Longships and back. The conditions, however, were otherwise very different, for, with the exception of the twelve hours of fine weather and smooth sea from noon till shortly after midnight on 6th November, it was throughout the average mid-Atlantic winter weather—namely, strong winds and resulting boisterous sea. Up till midnight on the 6th, *i.e.*, for 2,176 out of a total of 2,781 nautical miles, the mean speed works out at 24.65 knots; but, unfortunately, early on the 7th the wind freshened, gradually increasing to a furious south-west gale, which reached its height about 4 P.M., and reduced the average speed for the last twenty-four hours below 23 knots, and thus brought down the mean average for the completed voyage to 24.25 knots. Table VIII., giving the mean average speeds at the different stages of the voyage, shows very clearly the effect of this gale, unfortunate so far as

preventing the vessel from complying with the contract conditions, but giving those connected with the ship an opportunity of thoroughly satisfying themselves as to her behaviour when driving through the huge waves at about 22½ knots, without any racing of engine or sign of labouring, and dispelling the idea, current in some minds, that turbine-propelled ships do not show to advantage in heavy weather.

"The following particulars of the steam consumption are given in ion with the figures of coal consumption set forth in Table hroughout the voyage a careful record of the feed pump gave an average of 998,000 lbs. of water pumped into the er hour. Of this, about 114,000 lbs. was used by auxiliary ry exhausting into the feed heaters, 26,000 lbs. by the evapor- int supplying feed make-up and washing water, and about ;. for steam to the thermotanks, galleys and pantries, both of tter figures are based on data obtained from tests carried out he vessel left the Clyde. Hence, taking the average shaft wer as 65,000, the steam consumption per shaft horse-power works out as follows:—

	Total Water.	Per Shaft Horse-power Hour.
ain turbines - - - - -	851,500 lbs. =	13.1 lbs.
ıxiliary machinery - - -	114,000 lbs. =	1.75 lbs.
'aporating plant and heating -	32,500 lbs. =	.5 lb.
	998,000 lbs.	15.35 lbs.

rage amount of coal burnt per hour for all purposes = 43½ tons.
evaporated per lb. of coal = 10.2 from a feed temperature of 196°.
Water evaporated per lb. of coal = 10.9 from and at 212°.
'oal for all purposes per shaft horse-power per hour = 1.5 lbs.
Coal per square foot of grate per hour = 24.1 lbs.

king a mean displacement of 36,000 tons, this represents at 24½ consumption of almost exactly 11 lbs. of coal per 100 nautical r ton of displacement. The coal used was half South Wales Yorkshire, practically the same as on the official trials."

4. Blade Tip Clearances taken Cold and Hot.

H.P. Turbine.

COLD.

Expansion.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
		.057	.042	.044
		.052	.044	.048
		.056	.050	.049
		.058	.071	.058

HOT.

		.048	.026	.036
		.046	.023	.036
		.048	.032	.037
		.053	.054	.050

P.L.P. Turbine.

COLD.

Rotor.		Cylinder.	
	Bottom.	Top.	Bottom.
	.078	.078	.065
	.081	.079	.066
	.092	.076	.068
	.093	.090	.083
	.106	.103	.098
	.108	.108	.105
	.108	.108	.106
	.108	.108	.108

HOT.

	.055	.069	.040
	.054	.070	.038
	.056	.058	.035
	.054	.068	.050
	.058	.073	.065
	.063	.084	.066
	.053	.075	.063
	.053	.060	.061

S.L.P. Turbine.

COLD.

n.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
	.059	.073	.060	.067
	.066	.077	.061	.073
	.078	.088	.074	.084
	.082	.089	.075	.086
	.092	.096	.086	.088
	.103	.103	.096	.095
	.103	.103	.097	.093
	.103	.103	.087	.103

HOT.

	.052	.049	.052	.045
	.056	.049	.052	.045
	.060	.051	.059	.047
	.067	.052	.058	.050
	.067	.055	.063	.048
	.085	.063	.069	.050
	.077	.055	.061	.045
	.056	.045	.045	.057

Port Astern Turbine.

COLD.

n.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
	.071	.071	.068	.078
	.072	.078	.079	.087
	.103	.107	.096	.102
	.103	.103	.099	.103
	.102	.106	.099	.103

HOT.

	.060	.050	.054	.057
	.044	.045	.047	.052
	.058	.070	.051	.066
	.055	.053	.049	.051
	.044	.034	.046	.030

Starboard Astern Turbine.

COLD.

Expansion.	Rotor.		Cylinder.	
	Top.	Bottom.	Top.	Bottom.
1	.067	.070	.066	.067
2	.080	.077	.065	.072
3	.103	.099	.090	.100
4	.103	.095	.086	.100
5	.103	.095	.082	.094

HOT.

1	.060	.047	.065	.044
2	.053	.047	.039	.040
3	.067	.060	.053	.060
4	.072	.049	.052	.050
5	.082	.035	.060	.030

REMARKS.—On referring to the foregoing clearance data it will be noticed that the expansion of the turbines and blades is by no means uniform through, as for example at the position of the L.P. "wing" blades, that is, the 6th, 7th, and 8th expansion, where the blades are all of equal height, the expansion is unequal. Expansions 6, 7, and 8 of the starboard L.P. rotor all give a clearance of .103 at top when cold, but when heated up the respective clearances are then .085, .077, and .056, giving a variation of $.085 - .056 = .029$, that is $\frac{29''}{1000}$ difference in the clearance with equal blade heights. Again referring to the same L.P. cylinder, the clearances at the last three expansions are when cold, at top, .096, .097, and .087, which on being heated up alter to .060, .061, and .045, giving a difference of $(.069 - .045) = .024$, or $\frac{.024''}{1000}$ between the blades of the 6th and 8th expansion. Again, it will be noticed that the greatest clearance difference due to heating up is evident in the astern turbines, which show as much as $\frac{72}{1000}$ (.072) decrease in the clearance at the 5th expansion of the rotor (bottom) of the port astern turbine. Generally, the decrease in tip clearance due to heating increases from forward aft in any turbine, owing chiefly to the increase of blade heights being in that direction.

5. Type—Fast Passenger Steamer.

10,000 Horse-Power (approximate).

	No. 1.			No. 2.			No. 3.			No. 4.		
Boiler Pressure.	180 lbs.			183 lbs.			180 lbs.			180 lbs.		
H.P. Initial Pressure.	168 „			165 „			165 „			165 „		
H.P. Terminal Pressure.	23 „			22 „			22½ „			22½ „		
		P.	S.		P.	S.		P.	S.		P.	S.
L.P. Initial.		22 lbs.	22 lbs.		21 lbs.	21 lbs.		21½ lbs.	21½ lbs.		22½ lbs.	22½ lbs.
Condenser Vacuum.		27 in.	26¾ in.		27 in.	27 in.		26¾ in.	26½ in.		27½ in.	27 in.
Astern Turbines Vacuum.		24¼ „	24 „		24½ „	24½ „		23½ „	23½ „		24½ „	24¼ „
	H.P.	P.	S.	H.P.	P.	S.	H.P.	P.	S.	H.P.	P.	S.
Revolutions per min.	464	484	484	451	464	463	469	490	491	458	477	479
Mean Revs. all Shafts.	477			459			483			471		
Speed.	22 knots (against tide).			22.5 knots.			23.3 knots.			22.5 knots.		
	H.P.	P.	S.	H.P.	P.	S.	H.P.	P.	S.	H.P.	P.	S.
Dummy clearance.	.028	.028	.020	.030	.028	.020	.030	.029	.021	.03	.028	.020

NOTE.—Propellers, 6 ft. 6 in. diameter, 6 ft. pitch, mean slip, 18 per cent.

Gland Pressures.

H.P.		L.P.	
	Leak-off.	Steam.	Leak-off (shut).
	0	6 lbs.	8 in. vac.

gland of L.P. similar to ahead end gland.

stern" the reverse turbine initial pressure
ead turbine $24\frac{1}{2}$ in. vacuum, the H.P. tur-
condenser $26\frac{1}{2}$ in. vacuum, and the mean
per minute. The L.P. dummy clearance

e following are a few particulars of this large
Messrs John Brown & Co., Clydebank, to the
:—

- - - -	678 feet.
- - - -	72 feet.
- - - -	195 lbs. per sq. in.
- - - -	13
- - - -	185 per minute.
ller - - -	14 feet.
- - - -	13 feet.
ower - -	21,000
- - -	21 knots (trial).
rotor - - -	138 in.

de on the Clyde the speed attained by the
ots.

tio of the propeller is equal to .9, and the
t .6 or thereabout.

ders.—The large Cunarders, "Lusitania" and
with four lines of shafting and four propellers,
turbines are six in number, an H.P. on each
.P. and reverse turbine on each inside shaft.
independent of the L.P. turbines, and are
thrust block between the two (see sketch).
8 in. in diameter without the blades, the
used being 22 in. The line shafting is
the rotor spindles 33 in. diameter in the

bearings, which are about 5 ft. in length. The boilers (cylindrical) are twenty-five in number, working at a pressure of 200 lbs. per sq. in. The speed attained has exceeded 26 knots. For complete data see *Engineering* of 2nd August 1907 and of 8th November 1907.

8. Record of Wear Down at Main Turbine Bearings.

NOTE.—The following figures refer to the wear down as measured by the "bridge gauge" (see page 210). The first column shows the actual gauge clearance when new, and the other column the subsequent increase of clearance due to wear down when tested one year later.

	Original Marking.	After One Year.
H.P. Forward	.033	.034 (tight)
H.P. Aft	.036	.035
Port L.P. Forward	.041	.044 (tight)
Port L.P. Aft	.04	.045
Starboard L.P. Forward	.037	.044
Starboard L.P. Aft	.032	.036

Tip Clearance.—To allow of rotor and blade expansion under heat a certain working clearance has to be allowed between the tips of the rotor blades and the inside surface of the casing, also between the tips of the casing blades and the outer surface of rotor drum. This of course works out as a certain per cent. of loss by leakage over blade tips, no useful work being done by the steam in passing through.

It is calculated that if the loss is 3 per cent. when the revolutions are 600 per minute, then the loss at 200 revolutions per minute will be 27 per cent., as the per cent. loss varies as the square of the revolutions, or, which is the same thing, as the square of the rotor diameter.

So that, allowing for the increased expansion due to larger drums, the loss increases by the square of the diameter, or in other words, by the area open to leakage.

The following typical examples of blade tip clearances and dummy clearances were taken from the turbines of a large channel steamer:—

9. BLADE TIP CLEARANCES.

Taken when cold.

H.P. Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.041	.041	.052	.043	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{15}{64}$	$\frac{7}{32}$
„ 2	.051	.055	.059	.049	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
„ 3	.063	.055	.070	.061	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{1}{4}$
„ 4	.049	.045	.054	.051	$\frac{23}{64}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{11}{32}$

Starboard L.P. Turbine.

Rotor Drum, 68 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.070	.080	.068	.070	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{13}{64}$	$\frac{17}{64}$
„ 2	.072	.085	.070	.085	$\frac{5}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$
„ 3	.078	.090	.082	.092	$\frac{11}{32}$	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{17}{64}$
„ 4	.082	.092	.082	.085	$\frac{3}{8}$	$\frac{11}{32}$	$\frac{5}{16}$	$\frac{3}{8}$
„ 5	.085	.093	.085	.088	$\frac{7}{16}$	$\frac{11}{32}$	$\frac{11}{32}$	$\frac{11}{32}$
„ 6	.095	.123	.098	.115	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{7}{16}$
„ 7	.102	.125	.105	.112	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{15}{32}$
„ 8	.102	115	.105	.113	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{1}{2}$

red when Cold.

(1) Gauge initial pressure at 5th expansion.
(2) „ 6th „
(3) „ 7th „
(4) „ 8th „

next expansion.

W casing, rotor, and brass blades, the latter exp end the tip clearance may decrease by only

[*To face page* 166.

Starboard Astern Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.055	.062	.068	.070	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{17}{32}$
„ 2	.065	.064	.080	.082	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{7}{32}$	$\frac{17}{32}$
„ 3	.098	.102	.122	.128	$\frac{3}{16}$	$\frac{17}{32}$	$\frac{3}{16}$	$\frac{17}{32}$
„ 4	.098	.100	.125	.110	$\frac{7}{32}$	$\frac{17}{32}$	$\frac{7}{32}$	$\frac{17}{32}$

	Cold Setting.	On Trial.
H.P. Dummy Clearance - -	$\frac{15}{1000}$	$\frac{30}{1000}$
S.L.P. „ „ - -	$\frac{36}{1000}$	$\frac{21}{1000}$
P.L.P. „ „ - -	$\frac{36}{1000}$	$\frac{27}{1000}$

It will be noted that the H.P. dummy clearance increased when heated up, and the L.P.'s *decreased* when heated up.

Observe that the reverse turbine longitudinal clearance is much more *aft* than forward, to allow of the heating up and expansion of the drum and casing, which takes effect on the after end of the turbine.

10. Blade Tip Clearance Top and Bottom

TAKEN COLD.

H.P. Turbine.

Expansion.	Cylinder.		Rotor.	
	Top.	Bottom.	Top.	Bottom.
No. 1 - - - -	.040	.050	.032	.053
„ 2 - - - -	.042	.056	.037	.057
„ 3 - - - -	.052	.068	.041	.060
„ 4 - - - -	.050	.061	.034	.057

12. The following data refer to a cross-channel steamer :—

Full Speed Ahead (all Turbines).

Speed	22 knots.
I.H.P.	6,500.
	160 lbs.
	140 ,,
	20 ,,
	20 ,,
	23 in. vacuum.
:rn)	23 ,, ,,
	24½ ,, ,,
	630 per minute (all shafts).
	4 ft. 6 in.
	21 per cent.

cations. — The following figures from actual
ea as to the usual pressures and vacuum carried
eamer of moderate power :—

Type—Channel Steamer.

	160 lbs.
l)	150 ,,
; (initial)	24 ,,
L.P. shafts	25 in. vacuum.
	27 ,, ,,
	650 (all shafts).

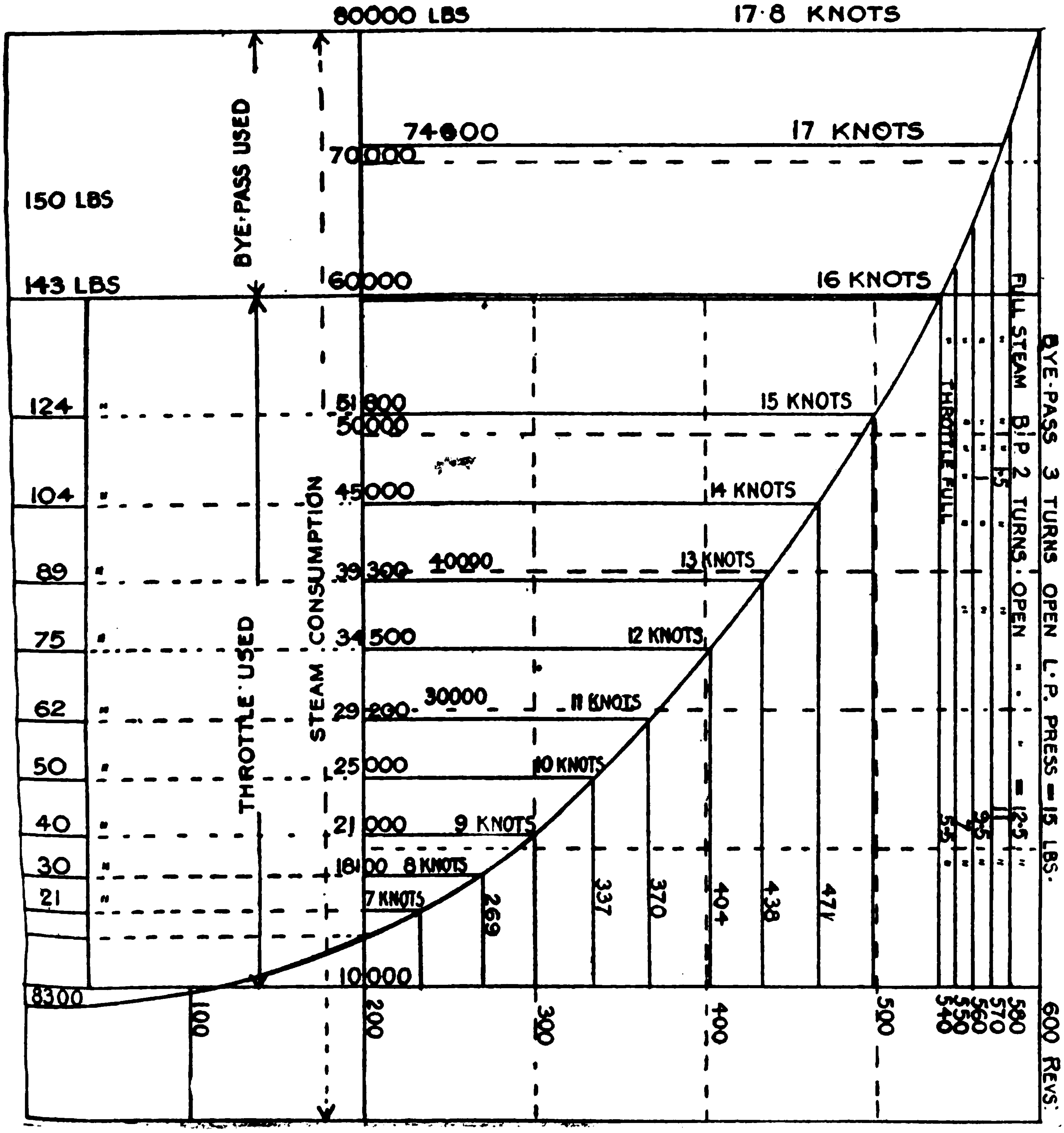
ıning ahead at reduced speed with *all turbines*
ressure is 80 lbs., L.P. pressures 15 lbs., and
um.

Type—Channel Steamer.

ıll Power Ahead with all Three Turbines.

	155 lbs. (gauge).	
ıre	146 ,, ,,	
ressure	13 ,, ,,	
ɔine pressure	12½ ,, ,,	
	27 in.	
ı gland pressure	1 lb.	These steam glands have no "leak-off" connections.
ıd pressure	1 ,,	
gland pressures	2 lbs.	
d pressures	3½ ,,	
e-power per hour	1.8 ,,	

:hat the L.P. turbine steam gland pressures are
turbine gland pressures, this being required to
ın of air into the turbine casing, which would
um.



Steam Consumption, Revolutions, and Speed Curve.

The above curve is developed from the steam consumption, revolutions, and speed of a turbine coastal steamer of about 5,000 horse-power.

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(14b.) Running Ahead with Two Outside (L.P.) Turbines only.

(Centre shaft running idly in vacuum at about ½ revolutions of outside shafts.)

Speed - - - - - - - - - -	15 knots.
H.P. turbine pressure - - - - - - -	20 in. vacuum.
L.P. turbine pressures, port and starboard - - -	20 lbs. (gauge).
Condenser vacuum - - - - - - - - -	27½ in.

(14c.) Running Full Speed Astern with Two Outside (L.P.) Turbines.

Speed - - - - - - - - - -	14 knots.
H.P. turbine pressure - - - - - - -	20 in. vacuum.
L.P. turbine pressures, port and starboard - - -	80 lbs. (gauge).
Condenser vacuum - - - - - - - - -	27 in.
Revolutions - - - - - - - - -	500.

Note.—When running ahead with all turbines one inch increase of vacuum gave 23 extra revolutions to each L.P. turbine shaft. The importance of a high degree of vacuum will thus be obvious.

In the steamer referred to above, the pitch of the outside propellers is 4 ft. 3 in., and of the H.P. (centre) 4 ft. 6 in. This difference of pitch is accounted for by the fact that the centre or H.P. shaft runs at less revolutions than the outside shafts, and therefore necessitates a larger pitch to obtain the same screw and ship advance. An example will make this clear.

Speed - - - - - - - - - -	20 knots.
Slip - - - - - - - - - - -	28 per cent.
L.P. turbine revolutions - - - - - - -	662 per min.
H.P. turbine revolutions - - - - - - -	620 „

To find the required pitch of propellers:—

$$\text{Knots} \times 6080 = \text{Pitch} \times \text{Revolutions} \times 60 \times \text{Effective \% advance.}$$

$$\text{Therefore, } 20 \times 6080 = \text{Pitch} \times 662 \times 60 \times .72,$$

$$\text{and, } \frac{20 \times 6080}{662 \times 60 \times .72} = 4.2 \text{ feet pitch (outside propellers),}$$

$$\text{and, } \frac{20 \times 6080}{620 \times 60 \times .72} = 4.5 \text{ feet pitch (centre propeller).}$$

Note.—$100 - 28 = 72$, and $\frac{72}{100} = .72$ effective advance of screw.

15. Consumption.—The consumption for this steamer worked out as 1.8 lb. of coal per *shaft horse-power* per hour, the power being calculated by means of the Denny & Johnson torsion meter (see page 262). This consumption, if reduced to the corresponding indicated horse-power, would be less per horse-power in proportion, as the shaft or transmitted horse-power is obviously less than what the indicated horse-power would be, were it possible to indicate the power.

The slip at full ahead speed worked out as 30 per cent. which, at ›eed of from 10 to 14 knots, fell off to 11 per cent.

Gland" Connections.—Sometimes this cock, fitted to the turbine casing at the "steam gland" port, or "outer of the two-way type, and is arranged either to (1) give ı to the gland, or (2) connect the gland with the condenser. ›ssary in the case of the H.P. steam glands, as, when the shafts are working only, the H.P. turbine is then merely ›ertly in a vacuum, due to the water action on the screw. › conditions the cocks can be opened to the condenser, but [.P. is running ahead with full steam, the cocks can then .o admit a steam pressure to the glands.

-With the above arrangement the "leak-off" port and pipe (see page 123) are both omitted.

ı in Idle Turbines.—When running ahead with all three › reverse turbines are revolving in a vacuum of from 22 to when running ahead or reverse with the outside shafts ntre or H.P. turbine is then revolving in a vacuum also of In this case the centre propeller is caused to revolve by ›f the water on the blades, being forced round in the ahead the outside shafts are running ahead, and forced round in direction if the outside shafts are running astern at the

ınce of High Vacuum.—As before mentioned, a high ' vacuum is of considerable importance to a turbine, as ›pansions of steam may be taken to vary in proportion to of vacuum obtained.

›e $\begin{cases} \text{Initial pressure in H.P. turbine} = 140 \text{ lbs. gauge,} \\ \text{Vacuum in condenser} = 26 \text{ in.,} \end{cases}$

acuum at the last L.P. turbine row will probably be some- ıy 24 in., so that

$$24 \text{ in.} \div 2 = 12 \text{ lbs.,}$$

and $15 - 12 = 3$ lbs. absolute terminal pressure.

$(140 + 15) \div 3 = 51.6$ expansions of steam in all (by pressures).

the vacuum is increased to say 28 in. in condenser and P. turbine exhaust the result will be as follows:—

$$26 \text{ in.} \div 2 = 13 \text{ lbs.,}$$

and $15 - 13 = 2$ lbs. absolute terminal pressure.

$(140 + 15) \div 2 = 77.5$ expansions of steam in all (by pressures).

›regoing it will be evident that the economy on turbine ends greatly on the vacuum carried.

16. Type—Channel Steamer.

(9,000 Horse-Power.)

Speed Trials (Ahead).

Revolutions per Minute (mean of three shafts).	H.P. Turbine Pressure.	Port L.P. Ahead Turbine Pressure.	Starboard L.P. Ahead Turbine Pressure.	Speed in Knots.
290	10 lbs.	21 in. vacuum	21 in. vacuum	10.68
409	35 „	12 „ „	12 „ „	14.73
512	60 „	0	0	18.12
608	115 „	22 lbs.	22 lbs.	20.82
680	152 „	23 „	23 „	22

17. Astern Trials (Two L.P. Turbines only).

H.P. Turbine Pressure.	Port L.P. Turbine Pressure.	Starboard L.P. Turbine Pressure.
18 in. vacuum	82 lbs.	80 lbs.
25 „ „	50 „	50 „

NOTE.—Average revolutions 540 per minute, and mean astern speed 14.4 knots, at higher L.P. pressure shown above.

18. Turbines in Vacuum.—(1) When running ahead with all three turbines the reverse (L.P.) turbines are revolving inertly in a vacuum.

(2) When running ahead with the two outside turbines only, the H.P. turbine and the two reverse (L.P.) turbines are revolving in a vacuum.

(3) When running astern with the two outside turbines, the two L.P. ahead turbines and the H.P. turbine are revolving in a vacuum.

The average vacuum indications in a case noted by the writer were as follows:—

In No. (1), reverse turbine vacuum, 24 to 26 in.

„ (2), H.P. vacuum, 25 to 27 in.

(3), L.P. ahead turbines, 26 to 27½ in.

19. Steam Gland Pressures.—With the steam gland and least off arrangements as formerly fitted at each end of the turbines, the pressures indicated, when running ahead full power, were as follows:—

	Outer Pocket (Steam Inlet).	Inner Pocket (Leak-off to L.P. 3rd Expansion).
H.P. Turbine (forward and aft glands) -	1 lb.	10 to 15 in. vacuum
L.P. Turbine (forward and aft glands) -	1 „	5 to 15 „ „

The "steam to glands" pipe is led off the main steam pipe, and is reduced in pressure by throttling the cock or valve to about 40 lbs.; this is, of course, still further reduced at the "outer pocket" of the glands by wire-drawing at the cock on the gland itself.

The inner pocket of the steam glands leaks off the small quantity of steam which finds its way past the brass rings and collars to the 3rd expansion of the L.P. turbines. All the leak-off connections from all three turbines forward and aft are connected up in this way when running ahead with all three turbines. When running ahead or astern with the two outer (L.P.) turbines only, the H.P. leak-off cocks are changed over direct to the condenser with the pressure results shown below.

20. Running Ahead with Two L.P. Turbines only.

	Outer Pocket (open to Condenser).	Inner Pocket (Leak-off to L. P. 3rd Expansion).
H.P. Turbine (forward and aft glands) -	27 in. vacuum	5 lbs. steam pressure

NOTE.—If the speed is reduced when running ahead with all three turbines, the "outer pocket" steam inlet pressure increases by a few pounds, and the "inner pocket" leak-off vacuum also increases by a few inches.

It should also be noted that the H.P. steam gland cocks are of the "two-way" type, to allow of the change over to condenser when required; whereas the L.P. steam gland cocks are only "one-way" type. The "leak-off" cocks from all turbines are single-way cocks.

21. Turbine Dimensions.—The following are the principal turbine dimensions of a large cross-channel steamer of about 10,000

I.H.P., speed 22 knots, and revolutions about 500, the propeller pitch being 5 ft. 6 in.:—

	H.P. Rotor.	L.P. Rotor.	Reverse Rotor.
Diameter	48 in.	68 in.	48 in.
Length	68 „	90 „	58 „
Number of Expansions	4	8	4
Number of Blade Rows in each Expansion	12	6	12
Total number of Rows of Blades	48	48	48
Blade heights in each Expansion—			
1st Expansion	1⅜ in.	1⅜ in.	¾ in.
2nd „	2 „	2 „	1½ „
3rd „	2¾ „	2¾ „	3 „
4th „	4 „	4 „	3 „
5th „	...	5½ „	...
6th „	...	8 „	...
7th „	...	8 „	...
8th „	...	8 „	...

NOTE.—It should be noted that the casings will contain the same number of blade rows as the respective rotors, that is, 48 rows in each.

Coal Consumption.—Regarding the all-important question of coal consumption, results have proved that at low or moderate speeds the reciprocating engine burns less per I.H.P. per hour than the turbine, but at high or maximum speeds the reverse is the case, the turbine showing in some cases an economy of as much as 20 per cent. and more over the reciprocating engine.

At the present time deep-sea turbine steamers compare very favourably with the ordinary reciprocating engine in the matter of coal consumption, and for river or channel service the turbine steamers often run at a lower consumption than those of the reciprocating type. Cruisers of the "Indomitable" class show the remarkably low consumption at top speed of about 1.3 lbs. of coal per horse-power hour, and at reduced cruising speeds and powers of from 1.8 to 2.2 lbs. per horse-power hour. These results are much below reciprocating engine practice for similar vessels.

The most conclusive tests as showing the superiority of the turbine over the triple engine at high speeds were those carried out some time ago by order of the Admiralty in the sister vessels "Topaze," "Sapphire," "Diamond," and "Amethyst," which were all designed and constructed similar in every particular, with the difference that the "Topaze," "Sapphire," and "Diamond" were supplied with triple-expansion engines, and the "Amethyst" with turbines.

The displacement of each vessel was 3,000 tons, and the estimated I.H.P. required for a speed of 21.7 knots was 9,800; it is therefore only reasonable and correct to assume that the power required to drive each vessel would be equal for the same speed. Working on this basis the results shown indicate a decided advantage in coal consumption in the turbine-propelled "Amethyst," as compared with the triple-expansion engined "Topaze" at the higher speed, but the reverse at lower speeds. The clear gain in coal at the maximum speeds is quite remarkable and constitutes a strong argument in favour of turbines; at 14 knots the conditions are, so far as economy is concerned, more equal; but when the speed was increased to 18 knots, it was found that the consumption on board the "Amethyst" was something like 20 per cent. less; at 20 knots it was nearly 30 per cent. less; and at the higher speed the improvement was still greater. The influence of this economy on the radius of action is very marked; for instance, the turbine-propelled ship could, with her 750 tons of coal on board, steam 3,160 sea-miles at 20 knots, as compared with 2,140 miles by the cruisers fitted with the ordinary machinery.

22. Coal Consumption.—A few examples of coal consumption, noted by the writer from actual practice, are appended.

(A) I.H.P. 8,500, speed 22 knots, consumption about 6 tons per hour.

$$\text{Then, } \frac{6 \times 2240}{8500} = 1.58 \text{ lb. per hour per I.H.P.}$$

(B) Shaft horse-power (not I.H.P.) 6,500, speed 20.7 knots, consumption about 5.2 tons per hour.

$$\text{Then, } \frac{5.2 \times 2240}{6500} = 1.79 \text{ lb. per hour per shaft H.P.}$$

$$\text{and, } 1.79 \times .9 = 1.61 \text{ lbs. per I.H.P. per hour.}$$

(C)	I.H.P. - - - - - - -	9,000
	Speed - - - - - - -	22.8 knots.
	Mean revolutions - - - -	520 per minute.
	Boiler steam - - - - - -	175 lbs.
	H.P. turbine - - - - - -	140 „
	L.P. turbines - - - - - -	12 „
	Reverse turbines - - - -	22 in. vacuum.
	Condenser - - - - - - -	$27\frac{1}{2}$ „ „
	Consumption - - - - - -	6.67 tons per hour.

$$\text{Coal per I.H.P. per hour} = \frac{6.67 \times 2240}{9006} = 1.66 \text{ lb.}$$

See also pages No. 152, 155, 158, 171, 194, 195, 203, 205, 206, 207, 208.

Running Astern.—In running astern, the consumption increases above that required for ahead, as the astern turbines are naturally not so efficient as those designed for ahead work only.

23. Blade Dimensions.—The following dimensions, taken from the L.P. turbines of a large ocean-going steamer of about 12,000 I.H.P., will afford a fair idea of the proportions between the blade widths, blade heights, and blade tip clearances adopted in actual practice. The L.P. turbines consist of 8 expansions arranged as follows :—

L.P. Turbines.

Rotor Drum, 7 ft. 9 in. Diameter.

Expansion.	Blades.	Blade Tip Clearances (Cold).
1st	10 Rows of $\frac{3}{8}''$ blades $1\frac{3}{8}''$ high and $1\frac{1}{8}''$ pitch - -	.08″
2nd	10 " $\frac{3}{8}''$ " $2''$ " $1\frac{1}{4}''$ " - -	.08″
3rd	10 " $\frac{3}{8}''$ " $2\frac{3}{4}''$ " $1\frac{3}{8}''$ " - -	.08″
4th	10 " $\frac{3}{8}''$ " $4''$ " $1\frac{1}{2}''$ " - -	.09″
5th	10 " $\frac{1}{2}''$ " $5\frac{1}{2}''$ " $1\frac{7}{8}''$ " - -	.10″
6th	10 " $\frac{1}{2}''$ " $7''$ " $2''$ " - -	.12″
7th	10 " $\frac{1}{2}''$ " $7''$ " $2''$ " - -	.12″
8th	10 " $\frac{1}{2}''$ " $7''$ " $2''$ " - -	.12″

Dummy clearance, .03″ (or $\frac{30}{1000}''$).

Observe that the blades of the 6th, 7th, and 8th expansions are all of the same height and pitch, the only difference being that of blade section, the 7th and 8th expansions having blades of a flatter surface section and a greater circumferential pitch, as the packing pieces are thicker.

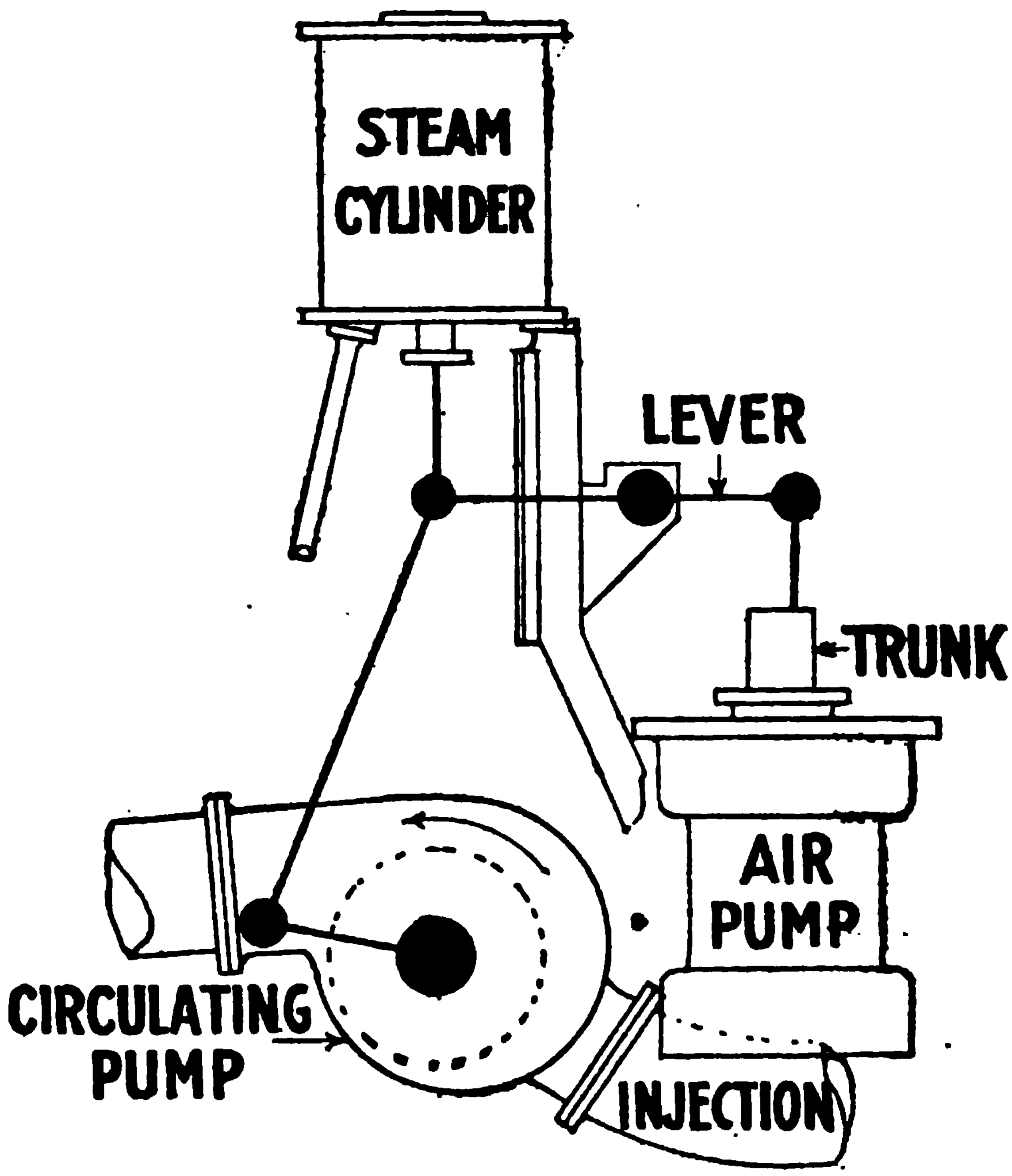
In steamers of from 5,000 to 8,000 estimated I.H.P. the blades vary in width from about $\frac{5}{16}$ in. in the first H.P. expansion to $\frac{1}{2}$ in. in the L.P. expansions, and in height from $1\frac{1}{16}$ in. in the H.P. turbine to about 8 in. or 10 in. in the L.P. turbines, depending on the diameter of rotor drum and the number of revolutions. In general, the larger the rotor drum diameter the shorter the blade heights.

Pumps.—All pumps fitted are of the independent double-acting type, and are as follows :—

Air Pumps.—The most recent practice consists of the fitting of independent twin air pumps of the Weir type, technically known as "wet" air pumps. These pumps draw as usual from the bottom of the condenser, and deliver the water into the feed tank and gravity

feed filter connections; from these the water is pumped by other independent donkey pumps into the surface feed heater and float tanks overhead, and from there the water is taken by the feed pumps proper and delivered direct into the boilers, purified and heated.

"**Dry**" **Air Pump.**—To obtain maximum efficiency in turbines, a high vacuum is of the utmost importance, and to produce this a special pump, known as a "dry air pump," has been recently introduced by Messrs G. & J. Weir, in addition to the ordinary independent air pumps as formerly fitted.



Air and Circulating Pumps.

The Weir "dry air pump" draws the air or vapour only from the condenser, and the ordinary air pump draws away the vapour still left and the condensed water. The dry air pump is therefore placed high up in position (usually above the circulating pump engine) with the suction branch from the condenser and the discharge pipe led away over the ship's side. By the aid of this special pump the vacuum carried has been as high as 29 inches. The dry air pump chamber is kept cool by means of a water jacket. Regarding the subject of vacuum the Hon. A. C. Parsons says: "An addition of 1 in. to the vacuum over 26 in. deducted 4 per cent. in the condenser from the steam used; a further increase of the same amount, 1 in.,

... Turbine. ... Khedive's Yacht "Mahrou..."

will thus be apparent that the number of independent service pumps required is in excess of those commonly fitted in engine-rooms of the reciprocating type.

Revolutions.—As previously stated, high revolution speed is necessary for the economical running of turbine engines. In the Clyde river turbine steamers the L.P. shafts revolve at about 800 revolutions per minute, and the H.P. shaft from 550 to 650 revolutions

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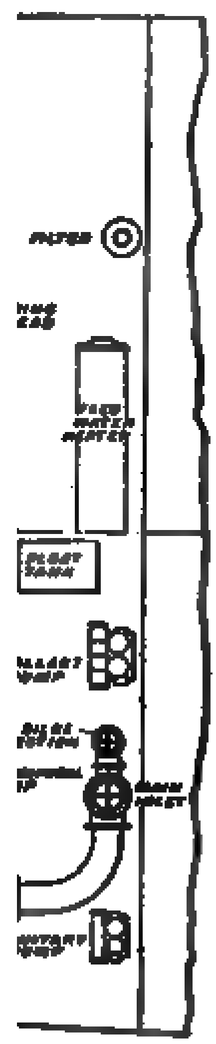

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known as a "Speed Indicator," or "Tachometer," which is fitted with a figured dial and pointer, and is specially employ[...] high revolution speeds in dynamos, &c. This instru[...] revolution in 10.

Reversing.—Contrary to the first impressions [...] pectations, turbine steamers have proved that [...] and reversing can easily be accomplished if the reve[...] made of sufficient power.

It should be stated, however, that sudden revers[...] the turbine, as excessive vibrations are set up by [...] propeller rotation.

Corrosion, Wear, &c.

Wear.—The wear on the brass blades of the t[...] tically *nil* after some years of service, the frictio[...] apparently having little or no effect on them. Wea[...] is very slight indeed, and only shows when something [...] It is understood that when running the turbines bal[...] all round, or in other words are floating more or less, [...] the wear to a great extent.

As mentioned previously, very little pressure bea[...] block, as the blades and dummy pistons together take [...] balance most of the propeller thrust.

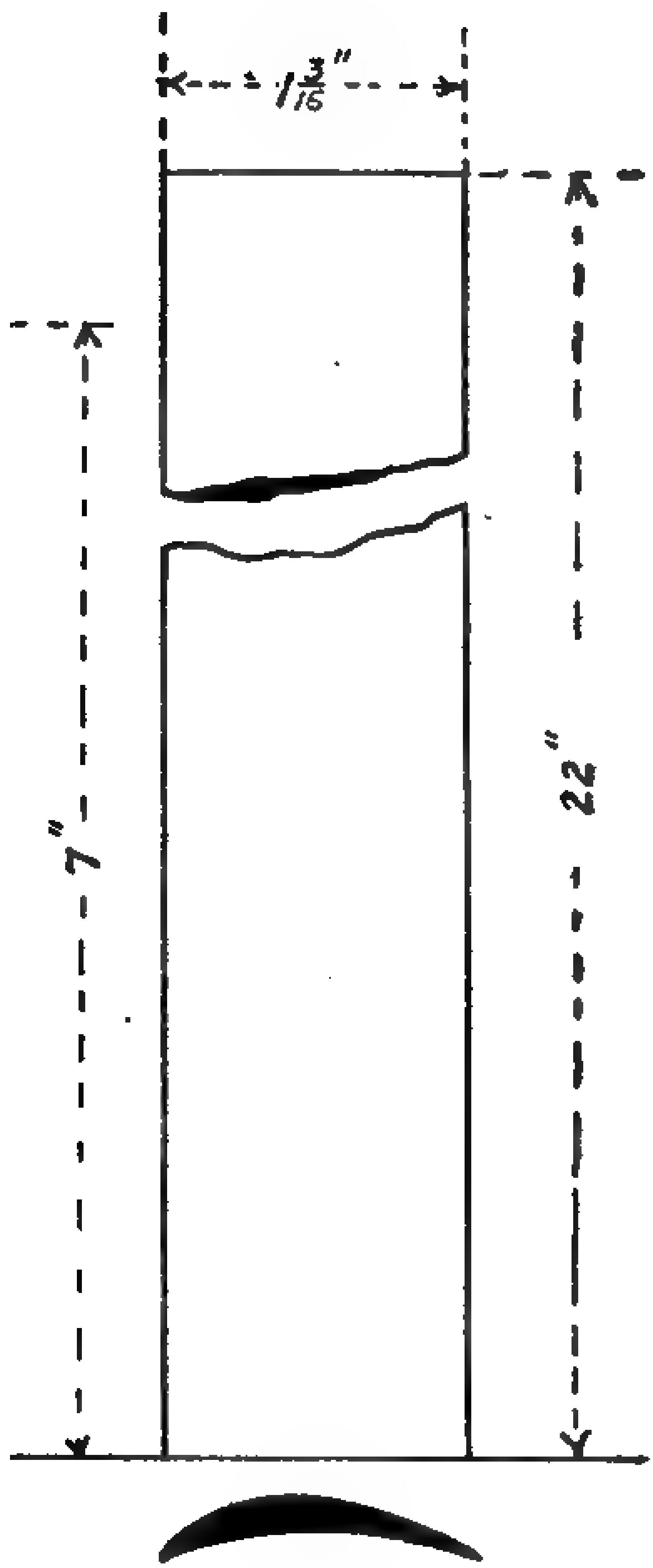
It is also evident that gyrostatic action of the roto[...] to be negligible altogether. "Whipping" is also e[...] to the very careful balancing to which the various pa[...] while under construction.

Wear of Gland Rings.—The loose brass rings [...] the steam gland packing occasionally show decide[...] after several months' service, and if not renewed so[...] are apt to give way altogether. A broken ring mig[...] damage to the blades, as the broken pieces might pa[...] drum and from there out again into the case and [...] of blades. This is quite possible, as the wheels or c[...] large openings cast in them which open up the dr[...] to the exhaust pressure of the turbine, and therefore t[...]

If one or two of the H.P. turbine gland rings [...] finally broke, the pieces might pass into the drum a[...] then out of the drum and into the L.P. turbine w[...] steam, and damage to the L.P. blades would be likel[...]

From the foregoing it will be obvious that the gl[...] be regularly renewed and kept up to their proper size[...]

Wear of Thrust Block.—After long service the t[...] and collars show practically no wear, the surface[...] polished, and in many cases the original tool marks [...] out. The wear, measured by the writer, in one cas[...] of an inch after two years' hard running.



d Packing Pieces (Full Size).

xpansion of a large Channel Steamer, the second large Ocean-going Steamer, and the third is from 'Express" Cunard Steamers "Lusitania" and

Owners—Messrs G. & J. Burns Ltd. *Builders*—The Fairfield Shipbuilding & Engineering Co. Ltd.

Low-Pressure Rotor, with Reverse Turbine Complete, R.M.S. "Viper."

View showing Thrust Collars, Main Bearings, Gland Collars, Ahead Dummy Piston, Ahead Turbine Expansions, Reverse Turbine Expansions, and Reverse Dummy Piston.

 [*To face page* 182.

Low-Pressure Rotor, in Lower Half Casing, R.M.S. "Viper."

View showing Ahead Steam Admission, Ahead Dummy Piston, Complete Ahead and Reverse Expansions. Also Reverse Dummy Piston.

 [*To face page* 182.

Wear of Bearings.—The wear of the bearings slight indeed, and in some cases is a negligible quantity, $\frac{2}{1000}$ or $\frac{3}{1000}$ either *down or up*, as sometimes the beari wear about equally all round, which circumstance would the rotor when running is in that condition known The wear down of the L.P. rotors in a case tested by only $\frac{4}{1000}$ of an inch after two years' service.

Wear of Blades.—As a general rule the turbine blac long service show no sign of wear whatever, the o noticed being the darkened colour produced by the heat of the steam, otherwise the blades appear u binding and brazing being as originally finished.

Machinery Vibration.—Absence of machinery vibra of considerable importance, especially in the case of steamer, and this advantage may be fairly laid claim engines, the machinery vibration being practically *nil*, propellers often set up severe vibrations aft just over the

R.M.S. "Viper" (see illustrations facing page 182 pelling machinery consists of three compound turbines, sure and two low pressure; each of these turbines driv shaft. The high-pressure turbine is on the centre shaft pressure turbines on the outer or wing shafts; with incorporated the reversing turbines which work in vacu ship is going ahead; the reversing turbines can st when going at full steam ahead in one and a half min time that the engineer receives the order from the capta

The vessel has four double-ended boilers 20 ft. 6 14 ft. in diameter, constructed for a working pressur per square inch. The air pumps and boiler feed-pump by Messrs G. & J. Weir Limited. The bunkers have 120 tons of coal.

NOTE.—On the trial runs over the measured mile attained a speed of fully 22 knots, with an air pressure water, while on the Cloch to Cumbrae and back conti mean speed was fully 21 knots, with natural draught an

Causes of Breakdown.—The principal causes of turbine machinery may be classed as follows:—

1. **Stripping of Dummy Rings.**—This may be by the (*a*) spindle bearings wearing down; or (*b*) the position longitudinally; (*c*) insufficient clearance when h

2. **Stripping of Blades.** — This may be brought (*a*) Wear down of spindle bearings. (*b*) Insufficient clea rotor blades and case, or between casing blades an heated up. It should be noted that the expansio

rotor when heated up is much more than that of the ', thus reducing the blade clearance considerably, and he risk of the rotor blades fouling the inside of the ng. (*c*) *Water hammer.*—If a body of water is introhe turbine casing through priming in the steam pipes, the e first and second expansion may be shattered by the ater-hammer" action.

e present there have been no cases recorded of "whipping" ·" of the turbine rotor which has resulted in stripped ›ugh this breakdown has been confidently predicted by ›ering experts.

›lute importance of the regular oil supply to each of ındle main bearings cannot be over-estimated, as most which have actually occurred of dummy ring and blade .ve been brought about by the temporary stoppage of ice, resulting, as may naturally be expected, in rapid ›f the main bearings, melting of the white metal, and wear down of the rotor, producing as a direct result the ›lades and stripping of some of the expansions. It is al practice to insert a sight feed glass in the oil supply able the engineer on watch to see that the oil service g order, and the supply pipes all clear.

›n in Rotor Drums. — One of the most serious t experienced in marine turbines is that of corrosion : rotor drums. This is undoubtedly due to moisture and n air introduced into the turbine either from the water re likely by leakage at the steam glands, which in the L.P. turbines aft draw in the air, if not blowing out steam. d be observed that aft the L.P. turbines work in a high acuum; the after steam glands thus may draw in air, ›ining with the cast iron, produces by chemical action ›n the inside of the cases. Again, the H.P. turbine runs n when the two outside turbines only are running, so this condition air may also be drawn into the turbine corrosion take place.

des of the rotors are sometimes painted over with com:heck the corrosion referred to.

ent the admission of air it is advisable to see that the l outer pocket cock is so regulated as to allow a constant ls of weak steam when the turbines are running.

f Dummy Rings.—In one or two cases it has been found ımmy rings have been worn away to some considerable he grinding action of chemical matter (iron oxide) proorrosion in the dummy cases. This may be assumed en brought about by galvanic action set up between ngs and cast-iron case when in contact in water, the atter thus formed probably grinding away the rings.

The water which accumulates is produced by condensation in the dummy grooves, and constitutes the so-called "water seal" of the dummy. The corrosion referred to is a matter of much importance, affecting as it does the mechanical balance of the turbine, not to mention the resulting loss of economy.

Heating Up.—Previous to starting up a turbine the following precautions should be taken:—

1. Open all turbine casing drains, with "wet" air pumps working slow.

2. Admit steam (reduced) to outer pockets of steam glands until the gauge shows a pound or two pressure.

3. Test dummy clearances before heating up and after heating up (This is important.)

24. Type—Channel Steamer.

H.P. Turbine.—On the cover being lifted, after about four months' service, signs of rusting were visible at the dummy "leak-off" port, also in dummy case. After dummy rings slightly worn, evidently due to contact, the "undercut" being much reduced and the corrosion decreasing in extent from the after to the forward rings.

L.P. Turbines.—Oxide of iron formation at "outer pocket" steam space, probably due to air (oxygen) finding its way into the turbines when working at low pressures. The inner pocket showed no signs of corrosion. The gland cases were more or less grooved showing that the brass rings were travelling round with the spindles. Four of these rings were broken in two places, and others were worn very thin (see sketch).

Signs of corrosion were evident at the reverse dummy fins on each side of each rotor fin and case fin. Some of these fins showed signs of actual contact.

The starboard L.P. ahead showed, on two sets of blades, severe oxidation; the dummies also gave similar evidences of oxidation. The S.L.P. dummy rings had evidently been in actual contact with the grooves, as these grooves were slightly recessed sideways, and the brass dummy rings themselves were broken off at intervals, and required to be renewed in the works. Although the finger plate indicated a clearance of about $\frac{20}{1000}$ in. when running, the faces must have been in actual contact, the reading being incorrect. This false indication may have been due to unequal expansion, when heated up, of the rotor, and the forward end of the casing, to which the finger plate is pinned down.

25. Type—Fast Cross-Channel Steamer.

L.P. Turbines.—After four months' service, corrosion (iron oxide) was evident inside the dummy cases, particularly at fillet of flange,

also on the rotor drum near the reverse fins. In the cylinder signs of corrosion were evident between the blade rows and near the leak-off port.

arboard L.P. ahead dummy when tested showed a variation :e at each side of from $\frac{10}{1000}$ in. to $\frac{18}{1000}$ in. This was evidently distortion of the casing due to unequal expansion by heat. · job was to grind up the dummies in place. The thrust collars indicated no actual contact as the tool marks were y visible. The gland rings were slightly worn, some show- ll ridge near the outer circumference, thus indicating that ıad not been moving round with spindle, as is often the case, ıot intended to be so. The rings showed signs of red oxide ıich, however, was quite easily removed with the finger.

Turbine.—The H.P. turbine showed signs of corrosion on ıies, both casing and rotor. Oxide of iron deposits were

1. Pitting on Casing Dummy. 2. Pitting on Rotor Dummy.

fairly large quantities between the brass rings and at the the dummy piston grooves; at some positions the oxide l up the space between the brass rings. The corrosion was ent forward and diminished aft. In the opinion of the ıay yet become necessary to construct both dummy case and ırass to reduce the corrosion referred to, which may reason- nsidered of a serious nature and requiring a drastic remedy. foregoing case priming had been going on for some time nted for most of the corrosion described. It should also be that the "eccentric" dummy gauge and "poker" gauge ne stuck fast and unworkable, the point of the poker gauge ly corroded, which would, of course, upset the reading of ıce.

ıver lifted after twelve months' service. No sign of wear is on thrust or main bearings, but both L.P. turbines show ridently the effects of steam and water friction, as at the

initial end of the rotor drums on the 1st expansion the surface is polished aft of each rotor blade row; this shows at the 1st, 2nd, and 3rd rows of the rotor 1st expansion. The burnishing referred to appears to have been caused by the water particles contained in the steam admitted to the L.P. turbines. The polished surfaces strongly resembled the effects of metallic contact, which at the positions noted was absolutely impossible.

26. Wear-down of Main Bearings.

(After four months' running.)

Type—Cross-Channel Steamer (22 Knots).

Bearing.	Amount Down.
H.P. forward	.003 in.
H.P. aft	.002 in.
P.L.P. forward	.002 in.
P.L.P. aft	.001 in.
S.L.P. forward	.002 in.
S.L.P. aft	.002 in.

The maximum wear-down was only $\frac{3}{1000}$ in. (.003 in.) for one season's running.

Finger-Plate Readings.—The finger-plate readings are occasionally so indexed that when the thrust surfaces are in close contact or "hard up" the finger plate's reading is less than the actual dummy clearance, which arrangement provides a margin of safety to work on. The following readings from a cross-channel steamer will perhaps make this clear:—

S.L.P. Dummy.—Half-rings, .300 in. forward and .270 in. aft taken out; half-rings, .310 in. forward and .260 in. aft put in, which gave readings of .018 in. on finger plate and .024 in. on dummy (by actual measurement on dummy rings). This gives a safety margin of (.024 − .018) = .006 in. or $\frac{6}{1000}$ in. Therefore, when even a $\frac{1}{1000}$ in. feeler could not be inserted between the finger plate and shaft groove, the dummies would still be clear by $\frac{6}{1000}$ in. or .006 in.

P.L.P. Dummy.—Hard up on thrust, .018 in. clear on finger plate and .026 clear on dummy.

H.P. Turbine.—Hard up on thrust, .009 clear on finger plate and .012 clear on dummy.

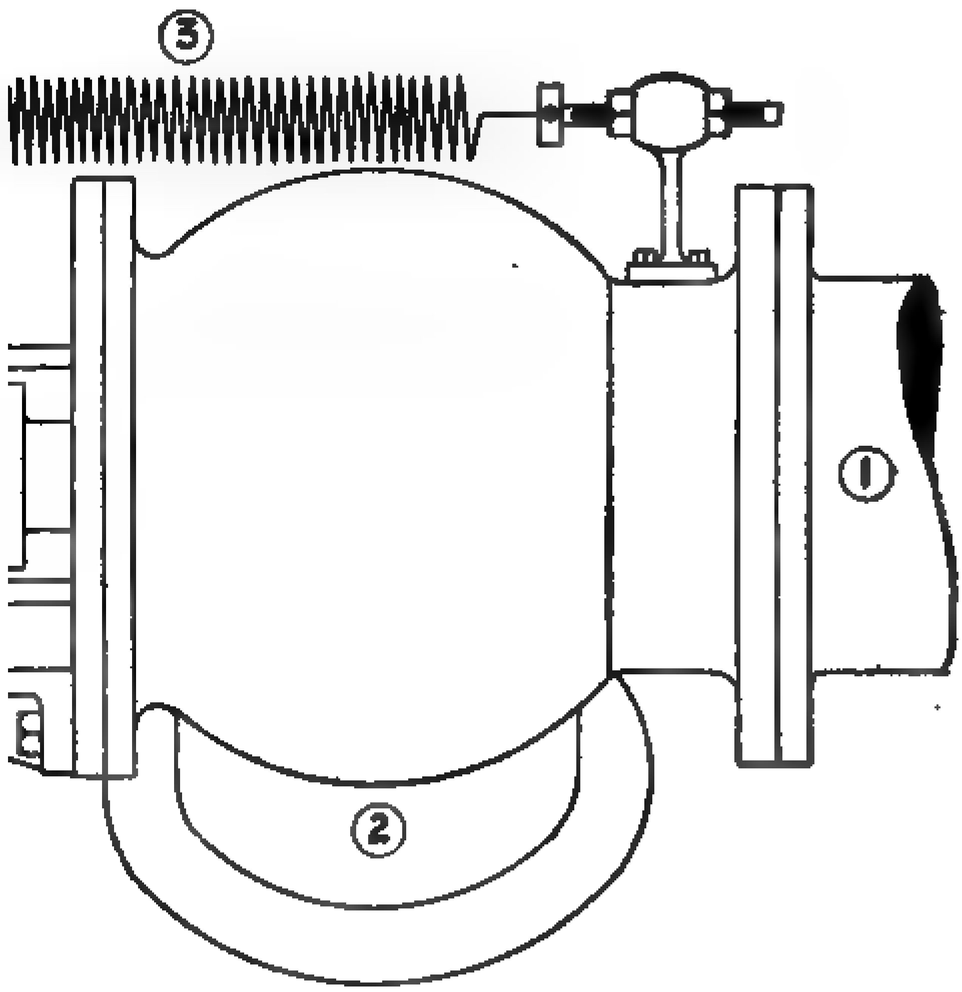
Propeller Vibration.—It has been found that the vibration aft, just over the propellers, is intensified—(1) When running in shallow water; (2) When rolling, on that side with the propeller nearest the surface of the water.

Steam Thrust and Propeller Thrust.—In the case of a fast channel steamer the L.P. dummy clearance varied about $\frac{8}{1000}$ in. or

ın initial pressure of 10 lbs. and 16 lbs., this difference the propeller thrust exceeding the steam pressure des at or under 10 lbs. pressure and the steam thrust of the propeller thrust at and over 16 lbs. steam

Working of Turbines.

1.—In going ahead full speed, steam is admitted am valve to the H.P. turbine, which it enters and



Spring-Loaded Non-Return Valve.

ıst from H.P. Turbine. 2. Admission to L.P. Turbine.
3. Pair of Springs.

l on the L.P. end of the H.P. exhaust pipes, and is intended to eam to the H.P. turbine when running with the L.P. turbines only.

the rings of guide and shaft blades alternately and t, exhausts at the other (after) end to the two L.P. either side.

L.P. turbine casings, the steam passes through the blades, and finally exhausts at a very low pressure sers. The independent air and circulating pumps ıum, and Weir's or other patent feed-pumps deliver to the boilers after it has passed through the feed-filter as commonly arranged. It should perhaps be

pointed out that the feed-heater fitted is often of the "surface" type —that is, one supplied with tubes through which the heating steam is blown, the water circulating through the chamber outside of the tubes. This arrangement prevents oily matter becoming mixed up with the feed-water when steam from the various auxiliary engines is used as heating steam for the heater.

Going Astern.—In going astern the main steam valve is closed and full-pressure steam is admitted by two valves to the astern turbines placed aft in the L.P. casings. When this is done, two large non-return valves, each fitted with a spring and placed between the H.P. and L.P. turbines, close automatically, and prevent the return of steam from the L.P. turbine back into the H.P. turbine.

The H.P. turbine is then in a vacuum, and the shaft and blades revolve idly as the propeller is set in motion by the water acting on the blades. Very little power is absorbed in this, as the vacuum offers but slight resistance to the rotation of the H.P. turbine blades.

Manœuvring.—As before stated, full-pressure steam can be admitted to the two L.P. turbines to drive ahead, the main steam being closed, or can be admitted to the two reverse turbines on the L.P. shaft to drive astern, the main steam valve still being closed, and the H.P. turbine revolving easily in a vacuum. It is also possible to give steam to one L.P. ahead turbine and the other L.P reverse turbine at the same time, to allow of quick turning of the vessel or for manœuvring at piers, &c.

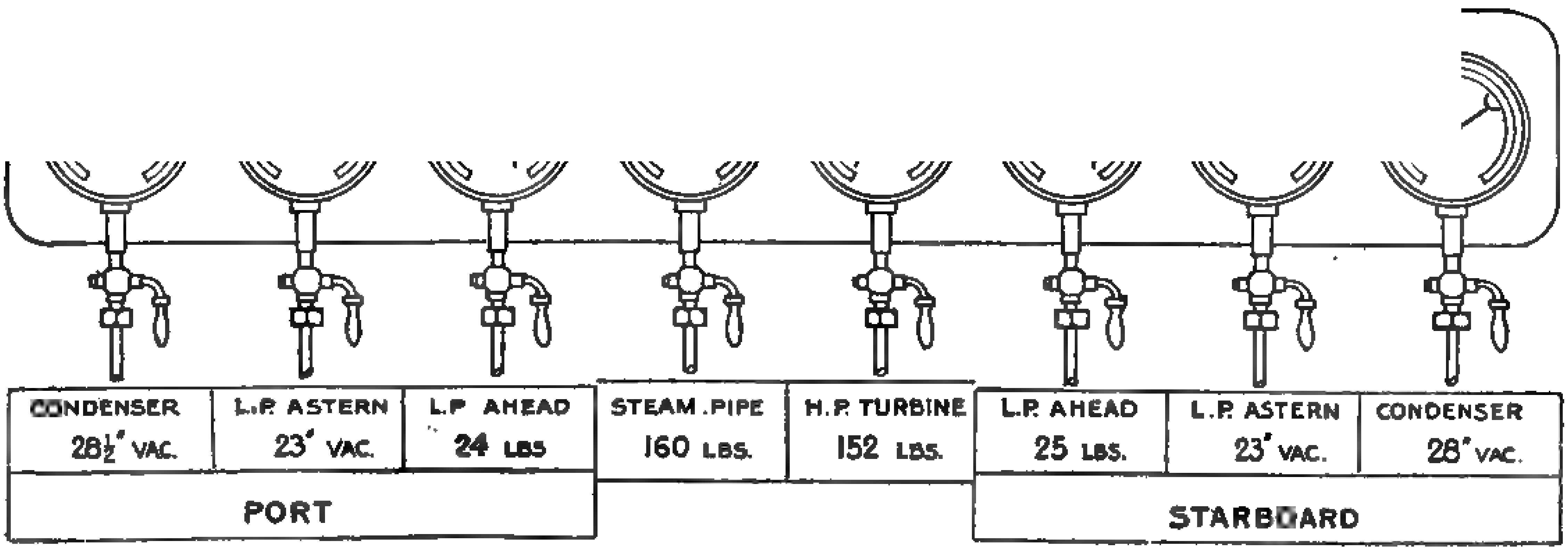
It will be obvious from the above that when going ahead with the two L.P. turbines, the L.P. reverse turbines are then running in a vacuum, and when going astern with the L.P. reverse turbines the L.P. ahead turbines are running in a vacuum.

Reversing.—Contrary to the first impressions and general expectations, the turbine steamers have proved that quick stopping and reversing can easily be accomplished if the reverse turbines are made of sufficient power. The "Dieppe" (cross-channel steamer) has supplied record figures for quick stopping and going astern, so that further doubt on this point is out of the question.

It should be stated that sudden reversing is very severe on the turbines, as excessive vibrations are set up by the change of rotation before the vessel comes to a stop.

Speed Regulation.—The engineer in charge regulates the speed as indexed by the telegraph, entirely by means of the revolution counters and the pressures shown on the various gauges and dials in connection with the turbine casings. Gauges are connected to the following points:—

1. Main steam pipe.
2. H.P. turbine.
3. Port L.P. turbine.
4. Starboard L.P. turbine.
5. Port L.P. astern turbine.
6. Starboard L.P. astern turbine.
7. Starboard condenser.
8. Port condenser.



Turbine Gauge Readings.

The above gives a fair idea as to the usual indications on the engine-room gauges of a large turbine steamer.

NOTE.—The gauges from No. 2 to No. 8 are of the "compound" type, and indicate either pressure or vacuum as required, according to the working of the various turbines.

Other gauges are fitted to show the oil pump pressure, cooling water pump pressure, &c. &c.

Number of Blades per Expansion.

In examining a list of blade numbers per row, or per expansion, it appears that, generally, the number of blades in each expansion of the casing is less than in each expansion of the rotor, although this does not hold good in all cases.

Case No. 1.—Type: Moderate Size Steamer.—The number of blades per row of the rotor in the first four expansions of the low-pressure turbines exceeds the number in the first four expansions of the cylinder, but from the 5th expansion on to the 8th expansion the number of blades per row in the cylinder exceeds the number in the rotor. In the H.P. and astern turbines the number of blades in the rotor exceeds the corresponding number in the cylinder throughout the four expansions of these turbines. Again, in the H.P. turbine the number of blades per row is the same for each expansion, but in the cylinder the number of blades per row *increases* from the 1st to the 4th expansions, the ratio of blade numbers being 1, 1.08, 1.16, and 1.2.

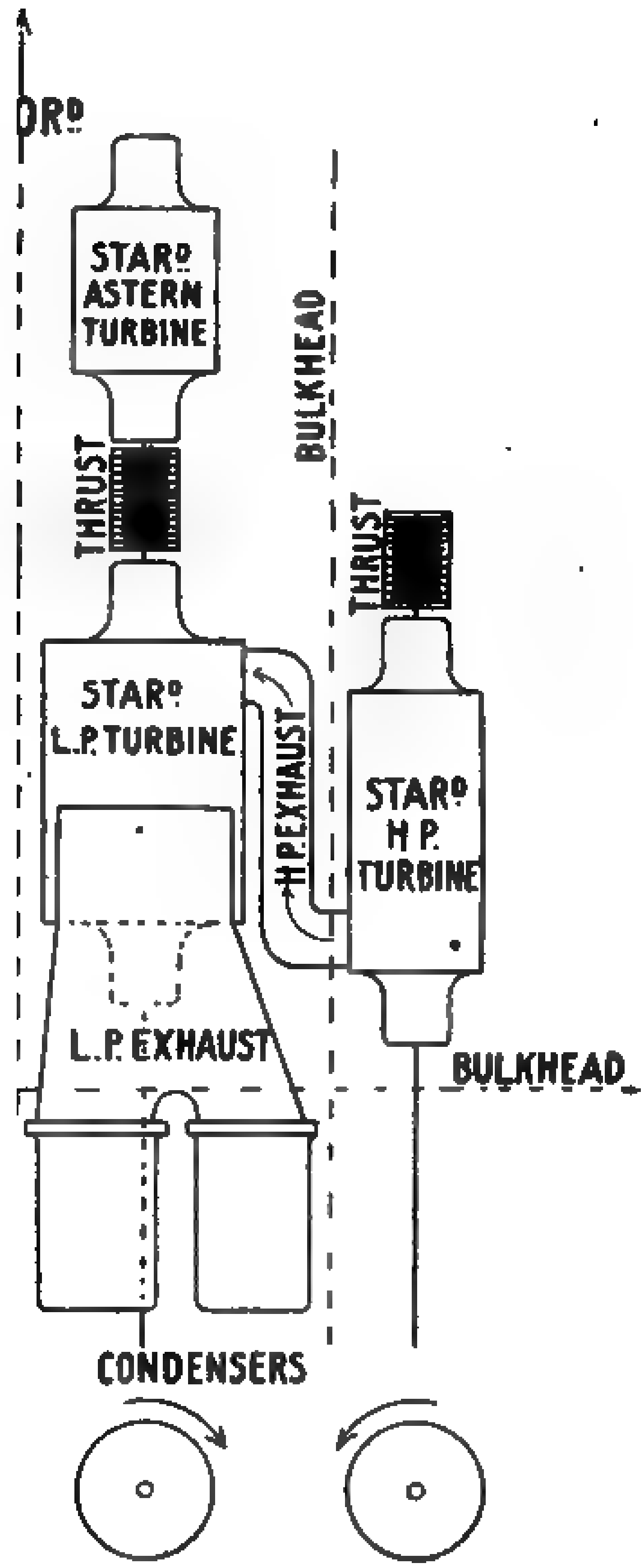
Referring to the astern turbines the number of blades per row is equal for the first six expansions of the rotor, but the 7th or last expansion has a smaller number. Again, in the astern cylinder the number of blades per row gradually increases from the 1st to the 6th expansion, the 7th expansion having a smaller number per row as in the case of the rotor.

Case No. 2.—Type: Small Steamer.—H.P. Turbine.—On referring to the H.P. turbine the number of blades per row in the rotor increases from the 1st expansion to the 3rd, and then drops away at the 4th, while in the cylinder or casing the number per row is constant for the 1st, 2nd, and 3rd expansions, and less for the 4th expansion. It is also noteworthy that the number of blades is less in the cylinder than in the rotor for the first three expansions but more for the 4th or last expansion.

L.P. Turbines.—In the rotor the number of blades per row increases from the 1st to the 3rd expansions. At the 4th expansion the number is much less, and from the 5th expansion to the 8th expansion the number gradually increases. In the cylinder the number of blades per .ow is constant for the first three expansions, then less in number and constant from the 4th expansion to the 7th expansion, the 8th or last expansion having a lesser number.

It is noteworthy that at all of the expansions except the 4th, the number per row in the cylinder is less than in the rotor, but at the 4th

Steam Turbine.



T.

'Lusitania" and "Mauretania."

e direction of shaft and propeller rotation,

d L.P. Right-hand Propellers.

t L.P. Left hand Propellers.

expansion the number in the cylinder exceeds that of the rotor in the proportion of 566 to 690 blades. This is exceptional, as generally the number of blades in the cylinder is less than the number in the rotor throughout all the expansions.

Astern Turbines—Rotor.—The blade number per row decreases from the 1st to the 2nd expansion, then increases in the 3rd expansion, but again falls away in the 4th expansion.

Cylinder.—The number per row is equal in the 1st and 2nd expansions (but less than the number in the rotor), and in the 3rd and 4th expansions the number decreases.

The number in the first two expansions of the cylinder is less than in the corresponding rotor expansions, but in the 3rd expansion of the cylinder the number is *more* than in the rotor, while at the 4th expansion the number is *equal* in rotor and cylinder.

Case No. 3.—Type: Large Fast Steamer.—Rotor.—The 1st and 2nd L.P. rotor expansions have equal numbers of blades, the 3rd expansion less, the 4th expansion a few more than the 3rd, the 5th and 6th expansions equal numbers of blades but each less than the 4th expansion, the 7th expansion less blades than the 6th, and the last or 8th expansion less blades again than the 7th.

Cylinder.—In the L.P. cylinder the first two expansions have equal numbers of blades, but the number is less than those of the corresponding rotor expansions. The 3rd expansion has less blades than the 2nd, the 4th more than the 3rd, the 5th and 6th less than the 4th, and the 7th and 8th expansions less blades still, the numbers falling gradually.

The last four expansions have blades of equal height but of increasing angle and therefore exit area. The angle of the last expansion is somewhere between 65 and 80 degrees. It is also worth pointing out that the 4th expansion, in addition to being fitted with a greater number of blades than the 3rd expansion, has, together with the 5th and 6th expansions, the greatest number of rows of blades per expansion.

27. Trial Results.

Runs.	Boiler Pressure.	H.P. Receiver Pressure.	P.L.P. Turbine Initial Pressure.	S.L.P. Turbine Initial Pressure.	Condensed Vacuum.		Vacuum in P.L.P. Astern Turbine.	Vacuum in S.L.P. Astern Turbine.
					P.	S.		
1	130	85	7	4	28″	28″	17	22
2	155	140	30	25	25	28	17	22

28. Gland Steam Pressures.

	H.P.	H.P. After End.	P.L.P. Forward End.	P.L.P. After End.	S.L.P. Forward End.	S.L.P. After End.
		7	3	[illegible]	3	3
		8	3	3	3	3

Consumption, Speed, and Power.

- - - - - - -	147 lbs.
- - - - - - -	120 „
ıum - - - - - -	28 in.
r minute - - - -	535
turbines per hour - -	45,300
auxiliaries per hour -	6,000
on per hour - - -	51,300
- - - - - - -	15 knots
- - - - - - -	3,000
· I.H.P. hour (all purposes) -	51,300 ÷ 3,000 = 17.1 lbs.

30. General Data.

-	147 lbs.
-	83 „
-	28 in.
-	1,831
-	13.5 knots
ı.	460
-	4,500 lbs.
ır	35,500 „
›ur	
-	2.25 „
-	17.5 „
lb.	
-	7.8

B.

Boiler steam - -	148 lbs.
H.P. turbine steam -	95 „
Condenser vacuum -	27.5 in.
Revolutions per minute { P. -	435
C. -	486.6
S. -	415.6
Speed - - -	13.5 knots
Coal per hour - -	4,840 lbs.

NOTE.—With H.P. steam shut off and steam admitted to L.P. turbines only at 15 lbs. gauge pressure, the speed was 12.25 knots, and the revolutions, P 469, and S 439 per minute. The coal consumed was 5,670 lbs. per hour.

31.

	A.	B.	C.	D.
Boiler steam -	145 lbs.	146 lbs.	147 lbs.	145 lbs.
H.P. turbine -	100 ,,	125 ,,	115 ,,	100 ,,
L.P. turbines -	0 ,,	3 ,,	3·5 ,,	1 ,,
Condenser vac.	27·5 in.	27.25 in.	26.25 in.	27.25 in.
Revs. (mean) -	490 per min.	480 per min.	492 per min.	455 per min.
Speed - -	14 knots	13·25 knots	13·75 knots	12.5 knots
Sea temperature	74 degs.	84 degs.	90 degs.	79 degs.

	E.	F.
Boiler steam - - - - - -	155 lbs.	155 lbs.
H.P. turbine - - - - - -	150 ,,	149 ,,
L.P. turbines - - - - - -	8 ,,	8 ,,
Condenser vacuum - - - - -	27 in.	26.75 in.
Revolutions (mean) - - - -	560 per min.	560 per min.
Speed - - - - - - -	15·5 knots	15·75 knots
Sea temperature - - - - -	78 degs.	82 degs.
Condenser temperature - - -	103 ,,	102 ,,
Steam consumption in one hour -	61,200 lbs.	59,400 lbs.
Coal burnt in one hour at an evaporation of 7.8 lbs. - - - -	7,846 ,,	...
Do. 8 lbs. - - - - -	...	7,425 lbs.

32. Test Runs showing effect of increase of Vacuum on Revolutions with constant Steam Consumption.

	H.P. Turbine Pressure.	L.P. Turbine Pressure.	Condenser Vacuum.	Sea Temperature.	Revolutions per Minute.			Mean.	Per cent. increase of Revolution Speed.
					H.P.	P.	S.		
	Lbs.	Lbs.	In.	Degrees.					
A	88	0	25	81	455	411	359	408	1·96 % for 1" vac.
B	88	½	26	81	454	424	369	416	
C	88	1	27	81	451	438	384	424	3·92 % for 2" vac.
D	145	4·5	25	78	560	484	473	506	4·2 % for 1¾" vac.
E	145	4·5	26	78	562	504	491	519	
F	145	4·5	26.5	78	562	509	499	523	
G	145	4·5	26.75	78	562	516	506	528	

NOTE.—In cases No. 29, 30, and 31 the form of the vessel was abnormal, and accounted chiefly for the low power efficiency and low speed shown.

33.

Boiler steam - - - - -	148 lbs.
- -	145 „
- -	6 „
- -	P. 26 in., S. 26½ in.
- -	86 degs.
- -	P. 112 degs., S. 108 degs.
- -	537 per minute.
- -	15.5 knots.
- -	23 per cent.
rance -	P. .013 in., C. .015 in., S. .025 in.

f Dummy Clearance, showing effect
d Port Propeller.

'	C.	S.	
n.	In.	In.	
18	.017	.019	... ...
25	.014	.018	Revolutions fell off 35 below normal.
26	.014	.018	... ...
17	.014	.019	... ...

at if the propeller falls off in propulsive
t as a drag on the hull, then the dummy
ne rotor position is now further aft than

aul.—"Starboard L.P. turbine opened
of 1st expansion found badly choked
f 2nd and 3rd expansions badly clogged
roots up to half height, but clear above
ated with dirty deposit. The cylinder
same condition as the rotor blades, but
st near blade tips, evidently indicating
ow outwards from the centre."
se priming had been going on for some

Dummy.—The starboard L.P. astern dummy radial fins were found burred on the edges more aft than forward; the burrs were removed and the "fin" edges recut. The blade tip clearances (taken cold) were as follows:—

35. Blade Clearances (taken cold).

Expansions.	1.	2.	3.	4.	5.	[illegible]
Rotor	.040 in.	.046 in.	.053 in.	.060 in.	.065 in.	.083 in
Cylinder	.031 "	.037 "	.047 "	.050 "	.055 "	.060 "

36. Data for Reduced Speeds.

	A.	B.
Boiler steam	185 lbs.	185 lbs.
H.P. turbine initial pressure	50 "	18 "
H.P. turbine terminal pressure	2½" vac.	17.5" vac.
P.L.P. initial pressure	3" "	18" "
S.L.P. initial pressure	3" "	20.5" "
L.P. astern turbines	26" "	S. 26.5" vac. P. 27" vac.
Condensers	28½" "	29" vac.

	A. H.P.	A. S.L.P.	A. P.L.P.	B. H.P.	B. S.L.P.	B. P.L.P.
Revolutions per minute	310	325	320	225	235	230
	15.5 knots.			11 knots.		
	H.P.	S.L.P.	P.L.P.	H.P.	S.L.P.	P.L.P.
	.012"	.026"	.025"	.014"	.030"	.032"

hat the dummy clearances increase with the

the full expansion due to heat acts on the metal as when actually running, whereas the L.P. turbines show an increased clearance when heating up whice is again decreased when running. This is due to the heavier material of the larger L.P turbines expanding more than the H.P., and acting to reduce the clearance by expanding the rotor in excess of the cylinder.

Case B.

	H.P.	P.L.P.	S.L.P.
	Inch.	Inch.	Inch.
Cold - - -	.026	.018	.025
Heating up -	.030	.031	.034
Running - -	.021	.021	.025

In the above case the dummy clearance increases in each turbine when heating up but decreases again when running.

Case C.—The variations in dummy clearance due to different conditions of working and running are shown in the following cases from actual practice:—

	Cold.	Heating up.	When Running.	Working with L.P. Turbines only.	Going Astern
	In.	In.	In.	In.	In.
H.P. Dummy clearance -	$\frac{50}{1000}$	$\frac{30}{1000}$	$\frac{20}{1000}$	$\frac{15}{1000}$	...
L.P. turbine dummy clearance	...	...	...	...	$\frac{40}{1000}$

Case D.

	H.P.	P.L.P.	S.L.P.
	In.	In.	In
-	$\frac{28}{1000}$	$\frac{45}{1000}$	$\frac{43}{1000}$
-	$\frac{17}{1000}$	$\frac{35}{1000}$	$\frac{32}{1000}$
-	$\frac{22}{1000}$	$\frac{40}{1000}$	$\frac{38}{1000}$
-	...	$\frac{46}{1000}$	$\frac{45}{1000}$

arances are slightly less than the above eed.

Turbine.

el Steamer.

pproximate).

a.

t. 6 in. Diam.)

Blade Heights.
1½ in.
2¼ "
3 "
4½ "

t. 9 in. Diam.)

Blade Heights.
1½ in.
2¼ "
3 "
4½ "
6 "
8 "
8 "
8 "

s "semi-wing blades."

Blade Heights.
1⅛ in.
2⅛ "
2⅞ "
2⅞ "
2⅞ "

s "wing blades."

aft of "leak-off" space, and

e.

ngs aft of "leak-off" space,

.

ed of starboard L.P. turbine due to the
ılades, as a remedy the *port* non-return
d port L.P. turbine was partly closed
t of raising the initial pressure in the
the revolutions of that turbine as the
s adjustment somewhat resembles the
g engine valve gear.

shutting port L.P. valve).

-	143 lbs.
-	8 ,,
-	P. 26¾ in., S. 27 in.
-	79 degs.
-	P. 564, C. 563, S. 499.
-	16 knots.
-	P. .023 in., C. .0125 in., S. .016 in.

shutting port L.P. valve).

-	150 lbs.
-	12.5 ,,
-	4 ,,
-	27 in.
-	79 degs.
-	S. 508, C. 536, P. 518.

o port L.P. turbine partly closed.

"**Expansions.**"—The following initial
I.P. turbine "expansions" should prove
ıich the pressures refer being of large

URBINE (No. 1).

- - -	210 lbs. gauge.
- - -	120 ,,
- - -	70 ,,
- - -	50 ,,
- - -	35 ,,
- - -	12 ,,
- - -	3 ,,
e) - - -	1 ,,

JRBINE (No. 2).

- - -	220 lbs. gauge.
- - -	125 ,,
- - -	80 ,,
- - -	53 ,,
- - -	33 ,,
- - -	16 ,,
- - -	6 ,,
e) - - -	2 ,,

For the foregoing the coal consumption varied from 2 lbs. per I.H.P. per hour at $\frac{1}{3}$ power to 1.3 lbs. per I.H.P. per hour at full power and speed, constituting a very fine performance indeed. The oil pressure was 8 lbs. per square inch at the bearings, and the oil temperature at discharge from bearings 133 degs., and when going through oil filter 50 degs.

43. Type—Pleasure Yacht.

Horse-power	5,200
Trial speed	17.75 knots.
Revolutions per minute	500 (designed).
Boiler pressure	150 lbs.
Propeller pitch	5 ft.
Propeller diameter	6 ft.

44. Turbine Data.

H.P. rotor, 42 in. diameter; L.P. rotor, 60 in. diameter; Reverse rotor, 42 in. diameter.

Expansions.	Blade Heights.		
	H.P.	L.P.	Reverse.
1	$\frac{9}{16}$ in.	$\frac{9}{16}$ in.	$\frac{3}{8}$ in.
2	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$\frac{3}{4}$ "
3	$1\frac{5}{8}$ "	$1\frac{5}{8}$ "	$1\frac{1}{4}$ "
4	$2\frac{1}{4}$ "	$2\frac{1}{4}$ "	$1\frac{1}{2}$ " *
5	...	$3\frac{1}{4}$ "	...
6	...	$4\frac{1}{2}$ "	...
7	...	$4\frac{1}{2}$ " *	..
8	...	$4\frac{1}{2}$ " *	...

* "Semi-wing" and "wing" blades.

:onsists of 16 blade rows (64 in all).
" " 8 " " (" ").

[.S. "Dreadnought."

taken from the pages of the *American*

—

/ilcox type, fitted for burning oil fuel in
ıl. Each boiler consists of 20 "elements."
sq. ft.
400 sq. ft.
5 to 1.
'e bars, 85 ft.

consumption at full power, 34,500 lbs. of coal per hour.
.rface of main condensers, 26,000 sq. ft.
.rface of auxiliary condensers, 6,000 sq. ft.
fts, 4
pellers, 4 (one per shaft).
hafts, 1 H.P. ahead turbine and 1 H.P. astern turbine.
shafts, 1 cruising turbine, 1 L.P. ahead turbine, and 1 astern turbine.

e wing propellers are placed forward of the centre propellers, as ia" and "Mauretania."

Turbines.

	Rotor Drums.	
	Diameter.	Length.
Cruising - -	5 ft. 8 in.	8 ft. $6\frac{3}{8}$ in.
H.P. ahead - -	5 ft. 8 in.	8 ft. $7\frac{1}{4}$ in.
H.P. astern - -	5 ft. 8 in.	3 ft. $1\frac{1}{2}$ in.
.P. ahead - -	7 ft. 8 in.	6 ft. 6 in.

Blading.

	H.P. Rotors.	L.P. Rotors.	
of blade rows - -	72	36	
of expansions - -	6	6	
le heights—			
st expansion - -	$\frac{7}{8}$ in.	$3\frac{7}{8}$ in.	
nd " - -	$1\frac{1}{4}$ "	$5\frac{3}{8}$ "	
rd " - -	$1\frac{3}{4}$ "	$7\frac{3}{4}$ "	
th " - -	$2\frac{3}{8}$ "	11 "	Wing-blades.
th " -	$3\frac{1}{2}$ "	11 "	
th " - -	5 "	11 "	

hould be noted that although the volume of steam entering es is fully more than that of the steam exhausting from the he blade heights drop from 5 in. in the last H.P. expansion to rst L.P. expansion, and this is accounted for by the larger herefore steam area annulus, of the L.P. turbines, which allow blade heights for similar area of steam flow. This difference accentuated in the case of the standard type three turbine two L.P. turbines being fitted only half the volume of steam the H.P. turbines is admitted to each L.P.

Trial Results.

...OURS' TRIAL AT 4,600 HORSE-POWER, 3RD AND 4TH OCTOBER 1906.

- -	220.0		220.0		220.0		220.0	
- -	0.58		0.58		0.60		0.56	
- -	Ten		Fifteen		Five		Means and totals	
	Starboard.	Port.	Starboard.	Port.	Starboard.	Port.	Starboard.	Port.
Steam pressure at engines, lbs.	210.0	210.0	210.0	210.0	210.0	210.0	210.0	210.0
Initial steam pressure in H.P. turbines, lbs.	34.5	30.2	25.5	22.8	31.6	28.0	29.5	26.1
,, vacuum in L.P. turbines, in.	13.3	16.1	9.6	12.6	11.4	15.4	11.1	14.2
,, steam pressure in cruising turbines	92.7	91.6	73.8	74.46	87.3	86.4	82.3	82.1
Vacuum in condensers, in.	27.85	27.75	27.73	27.48	27.3	27.48	27.7	27.57
Revolutions per min., inner shafts, number	197.94	194.76	215.82	207.89	205.82	196.95	208.2	201.69
,, ,, wing shafts, ,,	191.12	186.27	178.12	175.58	185.74	183.4	183.72	180.45
Horse-power, inner shafts	1,516.6	1,469.2	2,000.0	1,781.0	1,698.8	1,489.0	1,788.6	1,630.3
,, wing shafts	999.77	971.02	650.0	656.0	854.1	861.8	800.41	794.03
,, total	2,516.37	2,440.22	2,650.0	2,437.0	2,552.9	2,350.8	2,589.01	2,424.33
,, grand total	4,956.59		5,087.0		4,903.7		5,013.34	
Steam condensed per horse-power—								
Per hour from main engines, lbs.	20.8 } 27.2		25.1		27.3		25.9	
,, ,, auxiliary engines, lbs.	6.4							
Waste of feed water, tons	14.0		25.7		8.7 (gain)		31.0	
Water evaporated per lb. of coal, lbs.	10.3		10.28		9.98		10.2	
Coal per horse-power per hour, lbs.	2.6		2.42		2.74		2.6	
Exhaust steam from auxiliaries led into	Auxiliary condenser		On turbines		...		Various	
Speed of ship, by log, knots	13.0		13.0		13.0		13.0	

in.	...		...		0.13		0.30		0.80		0.80		0.50		0.75	
Duration of trial, hours	Two		One		Two		One		One		Two		One		Two	
	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.
Steam pressure at engines, lbs.	208	208	201	201	218	218	208	208	215	215	217	217	224	224	227	227
Initial steam pressure—																
High-pressure turbines, lbs.	4.0	1.0	10.0	13.5	16.0	15.5	14.0	13.5	74.0	69.0	74.7	68.2	87.0	82.0	87.0	82.0
Low-pressure turbines, lbs.	Vac. 17 in.	Vac. 22 in.	Vac. 16 in.	Vac. 20 in.	Vac. 14 in.	Vac. 18.5	Vac. 12 in.	Vac. 15.7	Vac. 5 in.	Vac. 7 in.	Vac. 5 in.	Vac. 4 in.	0	Vac. 3 in.	1.25	0.39
Cruising turbines, lbs.	25	25	25	25	55	55	55	55	179	178	179	179	180	178	180	179
Vacuum in condensers, in.	28.0	27.8	28.0	27.0	28.0	27.8	27.1	27.4	275	27.6	26.7	27.8	26.5	27.5	26.2	27.5
Revolutions per minute—																
Inner shafts, number	117.0	111.4	128.8	122.1	158.4	152.5	181.5	176.3	263.7	261.1	289.4	279.2	277.4	275.8	306.1	301.7
Wing shafts, number	121.7	115.6	140.8	138.6	155.4	149.2	154.8	149.7	249.6	245.0	245.9	242.2	266.3	273.3	264.0	262.5
Horse-power, inner shafts	415	327	485	394	842	844	1,291	1,197	3,542	3,465	4,858	4,388	3,901	4,383	5,631	5,505
,, wing shafts	279	282	436	433	567	518	485	448	2,213	2,081	1,957	1,889	2,697	2,767	2,327	2,412
,, total	694	609	921	827	1,409	1,362	1,778	1,645	5,755	5,546	6,815	6,277	6,598	7,150	7,958	7,917
,, grand total	1,303		1,748		2,771		3,423		11,301		13,092		13,748		15,875	
Steam condensed per H.P. per hour—																
From main engines, lbs.	34.9	58.5	37.4	41.6	38.2	49.7	32.3		16.01	19.9	17.7	18.16	15.3	18.9	16.6	
From auxiliary engines, lbs.	23.6		4.2		11.5				3.9		0.46		3.6			
Exhaust steam from auxiliary led into	Aux. cond.		On turbines.		Aux. cond.		On turbines.		Aux. turbines.		On turbines.		Aux. cond.		On turbines.	
Speed of ship, by log, knots	...		...		...		...		...		...		...		...	

Thirty Hours' Trial at 16,250 Horse-Power, 6th and 7th October 1906.

The first ten hours without closed exhaust; the next six hours with closed exhaust on turbines and auxiliary condenser; the last fourteen hours with closed exhaust on turbines and evaporators.

Mean Results.

Steam pressure in boilers	232 lbs.	
Air pressure in stokeholds	0.9 in.	
Duration of trial	30 hours.	
Initial steam pressures in—	Stbd.	Port.
High-pressure turbines	110.4 lbs.	109.3 lbs
Low-pressure turbines	0.6 „	3.75 „
Cruising turbines	...	...
Vacuum in condensers	27.5 in.	27.9 in.
Revolutions per minute, inner shafts	298.34	294.4
„ „ wing shafts	286.22	286.08
Horse-power inner shafts	5,092	4,988
„ wing shafts	3,350	3,500
„ total	8,442	8,488
„ grand total	16,930	
Steam condensed per horse-power	17.01 lbs.	
Waste of feed water	54.03 tons.	
Water evaporated per pound of coal	10.01 lbs.	
Speed of ship	19.3 knots.	

Measured Knot Trial, 9th October 1906.

Run.	Revolutions per Minute. Starboard. Inr.	Starboard. Wing.	Port. Inr.	Port. Wing.	Time along Knot.	Speed.	Horse-power. Stbd. all.	Port all.	All of all.
					Min. Sec.	Knots.			
1	354.9	324.8	342.0	320.8	3 10	21.78	13,774	14,125	27,899
2	353.2	333.3	351.4	328.2	3 13	21.45	13,663	13,670	27,333
3	348.5	329.5	347.3	325.8	3 10	21.78	13,140	13,388	26,728
4	340.8	328.1	340.8	327.4	3 13.5	21.39	12,842	13,270	26,112
			Simpson's mean			21.6		Mean	27,018

EIGHT HOURS' TRIAL AT 23,000 HORSE-POWER, ON 9TH OCTOBER 1906.

MEAN RESULTS.

Steam pressure in boilers	241 lbs.	
ssure	1.2 in.	
team pressures—	Stbd.	Port.
uising turbines	...	...
gh-pressure turbines	157.5 lbs.	154.5 lbs.
w-pressure turbines, pounds	7.7 „	12.0 „
1	27.0 in.	27.4 in.
ions per minute, inner shafts	337.2	333.3
„ wing shafts	322.2	321.7
ower, inner shafts	7,430	7,447
wing shafts	4,795	5,040
	12,225	12,487
te horse-power	24,712	
r indicated horse-power per hour	1.51 lbs.	
vaporated per pound of coal	10.03 „	
ss of feed water	25.1 tons	
er horse-power—		
engines per hour	14.41 } 15.56 lbs.	
iary engines per hour	1.15	
f ship	21.25 knots.	
19½ knots	21 per cent.	
full speed	26 „	

port on Working of Turbine Drains and Steam Glands.— from H.P. turbine to L.P. port turbine is opened when the e is stopped, and shut (except for one turn) when the H.P. unning. In port the H.P. drain is opened to bilges if air stopped.

rain on forward end of H.P. turbine has two connections, after end of the same turbine and one direct to the bilges. n after end is opened when heating up and shut when out of the pipes, &c.

heating up, steam is admitted to H.P. glands, but when full power this acts as a 'leak-off' to L.P. 3rd expansion. eds and powers it is required to shut the 'leak-off' con- l admit steam to the gland to prevent the formation of a d danger of air admission to the turbine. The pressure on s varied from 1 to 2 lbs., and on L.P. glands from 2 to 4

-In this steamer the dummy clearance cold was *less* than d up, whereas the reverse is generally the case.

Efficiency.—The over-all efficiency of a turbine may be as shown in the following case taken from actual practice :—

Data.

H.P. turbine, initial pressure - - - - -	173 lbs. (gauge).
L.P. turbines, terminal pressure - - -	26 in. vacuum.
Shaft horse-power - - - - - - -	17,560.
Coal per hour - - - - - - -	12.5 tons.
Speed - - - - - - - - -	$20\frac{1}{2}$ knots.
Evaporation (assumed) - - - - -	8.6.

Then,

Initial $\begin{cases} 173+15=188 \text{ lbs. absolute}=376.4 \text{ temperature Fahr.} \\ 376.4+461=837.4 \text{ absolute temperature.} \\ 188 \text{ lbs.}=847 \text{ B.T.U. latent heat.} \\ \text{Dryness}=1 \text{ (assumed).} \end{cases}$

Terminal $\begin{cases} 2 \text{ lbs. absolute}=126.3 \text{ deg. temperature Fahr.} \\ 126.3+461=587.3 \text{ absolute temperature.} \\ 2 \text{ lbs.}=1026 \text{ B.T.U. latent heat.} \\ \text{Dryness}=.778 \text{ (adiabatic expansion assumed).} \end{cases}$

$$\begin{aligned} \text{Heat drop} &= 847 \times 1 - 1026 \times .778 + 837 - 587.3 \\ &= 847 - 798.228 + 837.4 - 587.3 \\ &= 1684.4 - 1385.52 \\ &= 298.88 \text{ (say 299) B.T.U.} \end{aligned}$$

$$\text{Steam flow per minute} = \frac{12.5 \times 2240 \times 8.6}{60} = 4013 \text{ lbs.}$$

$$\text{Theoretical horse-power} = \frac{299 \times 778 \times 4013}{33000} = 28288 \text{ horse-power.}$$

Actual horse-power = 17560.

Therefore, $17560 \div 28288 = .62$ efficiency.

NOTE.—Steam consumption per H.P. hour $= \dfrac{33000 \times 60}{299 \times 778 \times .62} = 13.72$ lbs.

EXAMPLE.—Determine the theoretical heat drop at the 2nd H.P. expansion, the initial pressure being 112 lbs. (gauge) and the terminal pressure (or 3rd expansion initial pressure) 68 lbs. (gauge). Assume dryness factor of .97 and .95.

Then,

Initial $\begin{cases} 112+15=127 \text{ lbs. absolute}=345.4 \text{ temperature and } 870.7 \text{ latent heat.} \\ 345.4+461=806.4 \text{ absolute temperature. Dryness } .97 \text{ (assumed).} \end{cases}$

Terminal $\begin{cases} 68+15=83 \text{ lbs. absolute}=314.5 \text{ temperature and } 892.5 \text{ latent heat.} \\ 314.5+461=775.5 \text{ absolute temperature. Dryness } .95 \text{ (assumed).} \end{cases}$

$$\begin{aligned} \text{Then, } & 870.7 \times .97 - 892.5 \times .95 + 806.4 - 775.5 \\ &= 844.579 - 847.87 + 806.4 - 775.5 \\ &= 1650.97 - 1623.37 \\ &= 27.60 \text{ B.T.U.} \end{aligned}$$

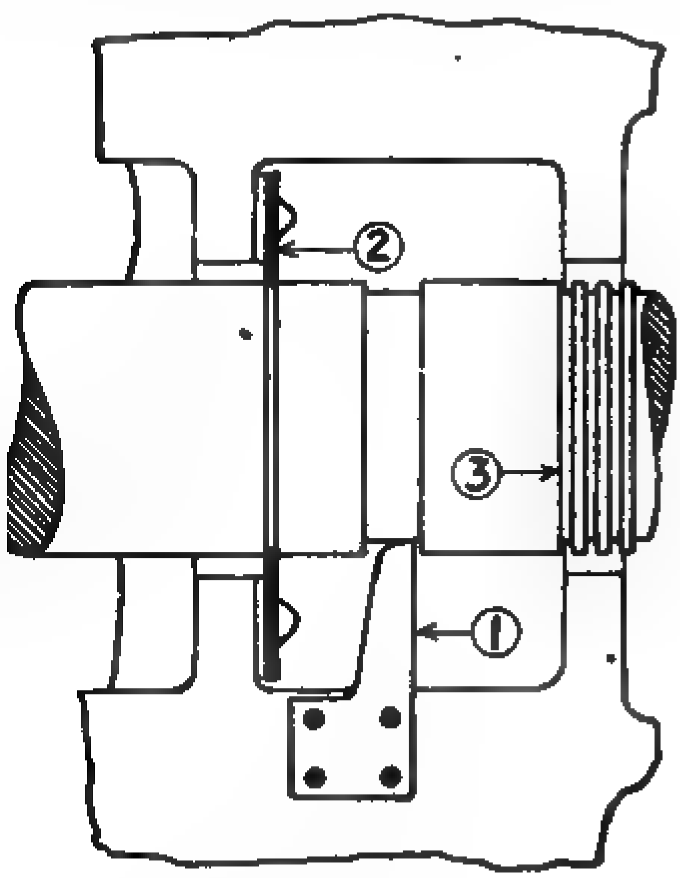
EXAMPLE.—Calculate the heat drop at the last L.P. expansion, the initial pressure being 22 in. vacuum (4 lbs. absolute) and the terminal pressure 26 in. vacuum (2 lbs. absolute). Dryness factors .8 and .79.

Initial $\begin{cases} 4 \text{ lbs. absolute}=153.1 \text{ temperature Fahr. and } 1006.8 \text{ latent heat.} \\ 153.1+461=614.1 \text{ absolute temperature. Dryness } .8 \end{cases}$

ıbsolute = 126.3 temperature Fahr. and 1025.8 latent heat.
⊦461 = 587.3 absolute temperature. Dryness .79.

$$006.8 \times .8 - 1025.8 \times .79 + 614.1 - 587.3$$
$$= 805.44 - 810.38 + 614.1 - 587.3$$
$$= 805.44 + 614.1 - 810.38 - 587.3$$
$$= 1419.54 - 1396.78$$
$$= 21.86 \text{ B.T.U.}$$

ırance Gauge.—A small flat plate, called a "finger lown to part of the lower casing frame at the forward ː of the plate projects against a collar on the turbine means of a feeler or wedge the clearance between



"Finger-plate" Gauge.

Plate. 2. Oil Deflector. 3. Steam Gland Rings.

ays be known. This clearance is noted when the ːted, and a record kept, so that any change of clear- · fore or aft can at once be noted by future testing. nge of position may occur through wear of the thrust to take up this (as mentioned elsewhere) a screw is half of the thrust block cover with an adjusting nut, gs fitted for adjustment of the lower or ahead portion set of these rings of varying thickness is supplied as an be inserted as required so as to maintain the same ː.

auges.—An appliance called a "bridge gauge" is of the rotor to test for possible wear down, which, as

will readily be understood, is a matter of paramount importance. The gauges merely consist of a piece of bent round bar iron with a centre boss through which is screwed and riveted a hardened steel pin with a flat end. The bridge piece is pinned down to the lower half casing just outside the gland (see sketch), and the pin is adjusted to a known clearance, say $\frac{50}{1000}$ of an inch, from the top of the shaft, so that by applying this test arrangement at intervals any wear down of the rotor can be detected by means of "feelers" placed between the flat end of the pin and the upper surface of the rotor spindle (see sketch).

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nly opened when the turbines are stopped and shut when ;ometimes, however, this drain is opened at regular intervals H.P. turbine of condensed water, but apparently this pre- not absolutely necessary, as careful tests, conducted on ıe writer, prove that only water actually condensed in the is blown out, as afterwards steam only is observed to blow 'rom this it is obvious that most of the steam condensed during the adiabatic expansion of the steam in the H.P. ses out with the exhaust steam to the L.P. turbines by way iver pipes, and does not remain in the turbine casing as been supposed. This matter is one of considerable practical , affecting, as it does, the safe running of the turbines. As lsewhere, the L.P. turbine drains are constantly open to pumps when running, but when the turbines are stopped ions marked up are: "When air pumps are stopped, switch ɔ bilges."

47. BLADING LIST.

(About 5,000 I.H.P. Speed, 16 knots.)

(NOTE.—The following figures are closely approximate.)

H.P. Rotor.

Diameter, 2 ft. 6 in.

nsion.	No. of Blade Rows per Expansion.	Blade Heights.	No. of Blades per Row.
		In.	
1	12	$\frac{3}{4}$	500
2	12	$1\frac{1}{4}$	500
3	13	$2\frac{1}{8}$	500
4	13	$3\frac{1}{2}$	500

H.P. Cylinder.

1	12	$\frac{3}{4}$	390
2	12	$1\frac{1}{4}$	420
3	13	$2\frac{1}{8}$	450
4	13	$3\frac{1}{2}$	460

Steam Turbine.

n Rotor.

r, 2 ft. 5 in.

's	Blade Heights.	No. of Blades per Row.
	In.	
	½	480
	11/16	480
	1	480
	1⅜	480
	2	480
	2¼	480
	2¾	470

Cylinder.

	½	380
	11/16	385
	1	390
	1⅜	395
	2	400
	2¼	410
	2¾	380

rings.

„

„ (radial fin type).

50. Reduced Speed Results.

A.			B.		
185 lbs.			185 lbs.		
50 "			18 "		
2½ in. vacuum			17½ in. vacuum		
3 "			20½ "		
3 "			18 "		
26 "			26½ "		
26 "			27 "		
28½ "			29 "		
S.L.P.	H.P.	P.L.P.	S.L.P.	H.P.	P.L.P.
325	310	320	235	225	230
	16 knots.			11 knots.	
S.L.P.	H.P.	P.L.P.	S.L.P.	H.P.	P.L.P.
.025	012	.026	.030	.014	.032

—In the new style of steam gland shown (page 124) to the four Ramsbottom rings which form the outer . The oil supply (under pressure) is led in through ee sketch) connecting the hood and flange of the rication thus arranged prevents undue wear of the lly gave trouble in the older type of packing, egular overhaul and renewal at intervals of from ths. At the same time it should be stated that nitting oil to the turbines is somewhat risky, and isiderable degree the advantage of this system of

ig Drains.—Small drain pipes fitted with brass the bottom of the ahead dummy casings to drain h accumulates there, and which tends to produce the brass strips and the cast iron of the casing or ' as described elsewhere.

ny drain is led to the exhaust pipe connecting the ines, and the L.P. ahead drains are led to the wet onnections at the after end of the L.P. turbines.

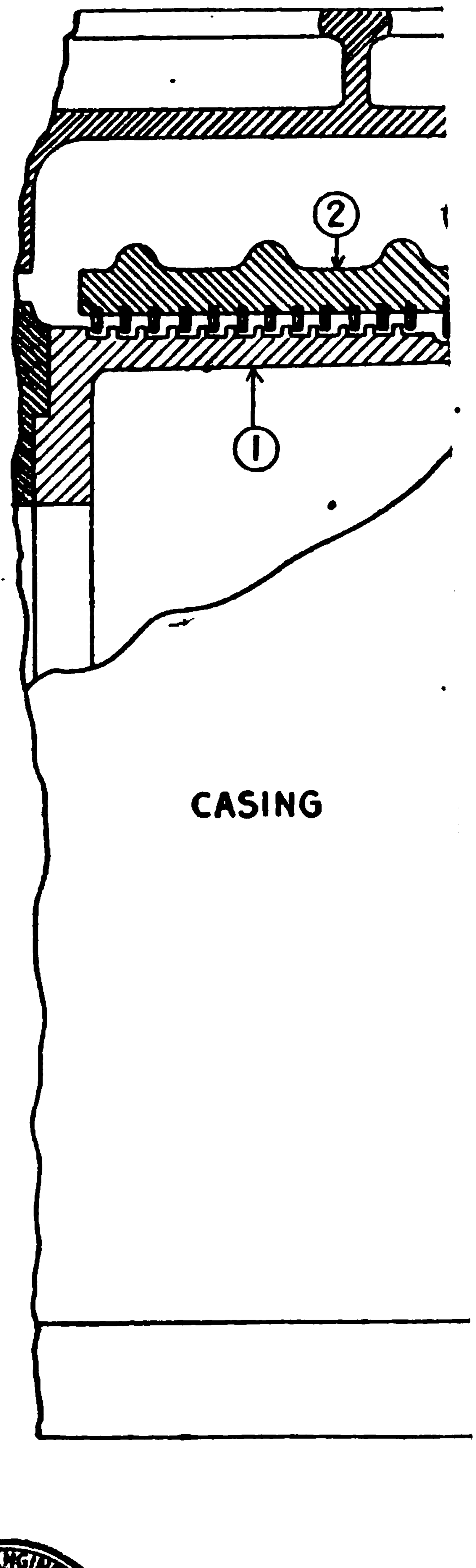
ns are also fitted to drain off the water when the air

.

immies drain off into the ahead casings by way of hrough the reverse dummy casings.

re, as a rule, only opened when the turbines are

ummies.—The dummies are sometimes drained ut in the brass rings of the dummy casing. These . wide, and are only cut in the lower half case near at is, at the lowest position. The condensed water the slots to the drain pipes fitted on the turbine



(1) Casing Dummy
(2) Rotor Dummy.
(3) Brass stop for r
(4) Micrometer gau

away clear of the handle, and a half-turn given to the spindle, the spindle end touches the actual end of the rotor, which, of course, is further aft than the stop; if now the wheel is screwed *back* against the spindle collar (8) the index on the wheel circumference and the fixed pointer will read the dummy clearance in thousandths of an inch. The spindle end coming into actual contact with the rapidly revolving rotor is apt to wear away, which would result in a false indication being registered. The length of the spindle, however, can always be tested by means of the stop-piece (1), which acts as a check on the length, and indicates 0 if no wear has taken place; if, however, the spindle end has been gradually ground away, the wheel will indicate the amount when screwed back against the collar and the spindle end in contact with the stop. The amount of wear thus measured would require to be deducted from the reading of the gauge when in contact with the rotor as shown in the sketches.

EXAMPLE.—The micrometer gauge when tested against the "stop" registers $\frac{5}{1000}$ in., and when testing for dummy clearance shows $\frac{30}{1000}$ in.; determine the actual dummy clearance.

Then, $\frac{30}{1000}$ in. $-\frac{5}{1000}$ in. $=\frac{25}{1000}$ in. or .025 in. actual clearance.

To take a Reading.—(*a*) With micrometer gauge spindle (2) in contact with stop (1) see if wheel (7), when screwed back against collar, reads 0 on index (10).

(*b*) Screw wheel back clear of collar, and give spindle one-half turn by handle to bring spindle end up against end of rotor.

(*c*) Screw wheel forward again against collar, and note reading of index, which will then be the dummy clearance in thousandths of an inch.

(*d*) Ease wheel again, and give spindle one-half turn to bring it back clear of rotor and into ordinary "out of gear" position against collar and stop.

Expansion forward of Casing Feet.—As previously explained, the turbine casings are bolted down securely aft, and are allowed to expand forward when heated up; this is arranged by the bolt holes in the forward feet being larger than the bolts. In a case measured the movement was as follows :—

H.P. casing expanded forward	.055 in.
P.L.P. casing expanded forward	.043 in.
S.L.P. casing expanded forward	.045 in.

PROPELLERS.

Turbine Propeller Balance.—Turbine propellers are carefully balanced on knife edges, a spindle being passed through the hole in the boss for that purpose when shop testing, and the difference in weight made up or taken away on the astern or forward surface of the blade. The thrusting surfaces of the blades are then carefully

-ed. the decided advantage obtained by

$^{1}\times .7854 = .3$ Projected Area Ratio.

aw

the blade. The thrusting surfaces of the blades are then carefully

chipped and highly polished, the decided advantage obtained by this finish in reducing skin friction on the blades being now generally

ɔropellers are always fitted with long tapering ·ater from the blades having a clear run aft, and ıo resistance to the thrusting column of water.

igh revolutions necessary to obtain suitable : pitch of the propellers is small. An ex- :lear.

17 knots, revolutions 400, and slip (assumed) required pitch of the turbine propellers.

n, $\frac{17 \times 6080}{400 \times 60 \times .78} = 5.5$ feet Pitch.

78 per cent., and $\frac{78}{100} = .78$ efficiency of advance.

ıe propellers is also necessarily much less than :d space aft, and to obtain the required blade io of blade surface to disc surface is developed. ›lains the somewhat peculiar shape of blade jority of turbine propellers (see sketches), all roader in proportion to length than ordinary

ınd necessary during the period of the steamer's : propeller blades, giving more or less surface results to be beneficial, before the required d. This indicates that calculations relating to ot always give the results desired in practice, ıeed for still more reliable information as to marine screw propeller, and, if at all possible, ethod of accurate design, which will obviate · "trial and error" experiments with blades of ıd contour.

ordinary marine reciprocating practice the ratio," or pitch divided by the diameter, gives ›m 1 to 1.4, but in nearly all turbine practice : out as a decimal figure, such as .8 or .9; in is less than the diameter instead of being more.

›itch of an ordinary propeller is 15 ft. and the ıe "pitch ratio."

ıen, $15 \div 12 = 1.25$ Pitch Ratio.

pitch of a turbine propeller is 8 ft. and the "pitch ratio."

Then, $8 \div 9 = .88$ Pitch Ratio.

liameter of an ordinary propeller is 12 ft. and ea 34 sq. ft., find the "projected area ratio."

$\div 12^2 \times .7854 = .3$ Projected Area Ratio.

EXAMPLE.—The diameter of a turbine propeller is 5 ft. and the projected blade area 12 sq. ft., find the "projected area ratio."

Then, $12 \div 5^2 \times .7854 = .6$ Ratio.

NOTE.—The *projected* blade area is the actual area of blades as aft looking forward, and constitutes the effective "thrusting the propeller. The "projected area ratio" is required in design calculations, and varies from about .45 to .60 of ea.

ent Slip per cent.—In turbine steamers the apparent slip tly with speed variation and weather conditions. A typical en below.

LE.—At a speed of 21 knots the revolutions of the turbines el steamer are 635 per minute; to find the apparent slip f the propeller pitch is 4 ft. 6 in.

ULE.— $\frac{\text{Pitch} \times \text{Revolutions} \times 60}{6080} = \text{propeller speed.}$

Therefore, $\frac{4.5 \times 635 \times 60}{6080} = 28.2$ knots.

Then, $28.2 - 21 = 7.2$ knots apparent slip.

And, $\frac{7.2 \times 100 \text{ per cent.}}{28.2} = 25.3$ per cent. slip (apparent).

Variation.—Turbine propeller blades generally present true aces, as no decided gain or advantage has been discovered y the adoption of pitch variation, either radially or peri- the all-important factors in propeller design consisting of t adjustment of pitch, diameter, and surface. At the same hould be mentioned that the propeller blades of the " Cunarders were originally constructed with a peripheral ation, the mean pitch being somewhere about 16 ft. This wn by the following: Speed 25 knots, revolutions 190, slip 15 per cent.

Then, Pitch $= \frac{25 \times 6080}{190 \times 60 \times .85} = 16$ feet.

ller Efficiency.—Above a certain revolution speed a pro- given diameter, pitch ratio, and area ratio rapidly loses cy as cavitation sets in and reduces the effective thrust. atio therefore increases, which produces a correspondingly opeller efficiency: it is thus impossible to obtain a maximum fficiency in combination with a maximum turbine efficiency. of comparatively small diameter also possess the dis- of losing in efficiency against head seas and head winds, in results in increased slip ratio. The problem is then to oth propeller and turbine speeds so as to give the highest *mbined* efficiency It will have been observed that, particu- rbine steamers, the actual sea speed often falls away from

Turbine Propeller.

Pitch, 5 ft. ; Diameter, 6 ft. ; Pitch Ratio = .83.

(Khedive's Yacht.)

Messrs A. & J. Inglis & Co. Limited.

Propellers of Turbine Steamer.

(Khedive's Yacht.)

Messrs A. & J. Inglis & Co. Limited.

the trial trip speed, and the reason of this will perhaps be understood from the foregoing explanation.

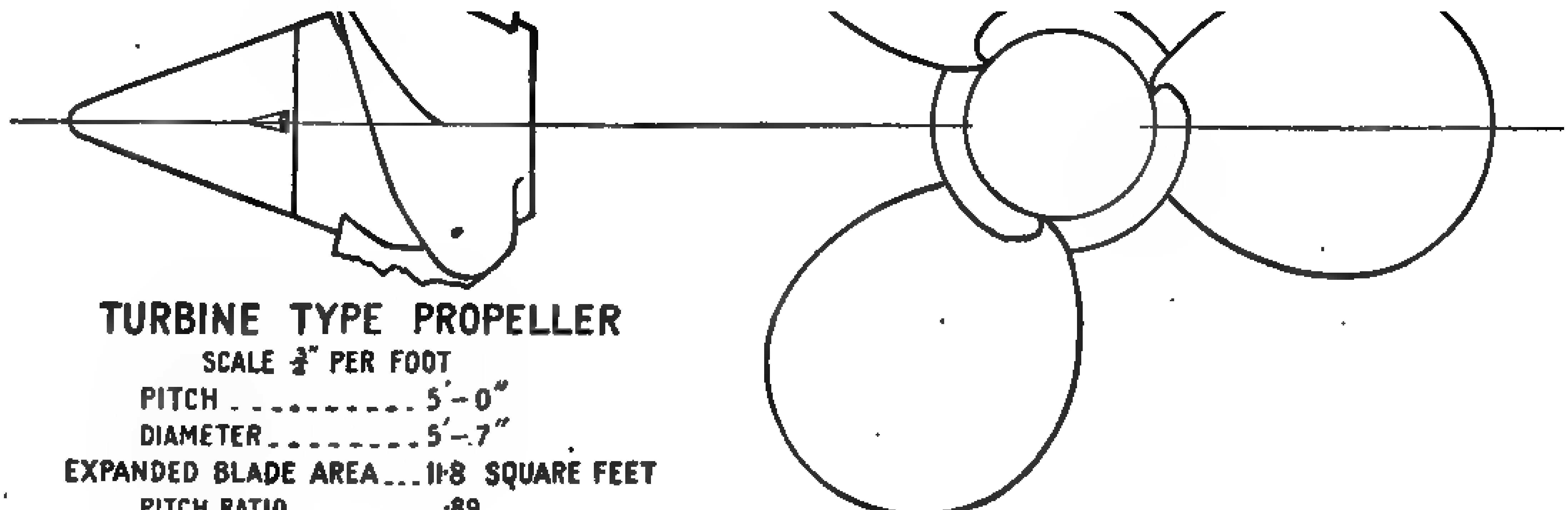
Cavitation.—Cavitation is caused by the ineffectiveness of the atmospheric pressure to press up the water at the back of the blades fast enough to allow of effective thrust. This usually occurs at high revolution speeds, and at high blade pressures per square inch.

Regarding cavitation and slip, Mr E. M. Speakman says :—

"Cavitation is partly the result of attempting to obtain too much work per square foot of blade area, and partly of excessive peripheral speeds. It has been found, by bitter experience occasionally, that there is a narrow limit to the tensional pressure possible on the water, per unit of projected area, beyond which the propeller efficiency drops very rapidly. This pressure is approximately from 10 to 12 lbs. per square inch at a depth of 12 inches below the surface, and to reduce the total thrust to this, sufficient blade area must be provided, which, in conjunction with certain practical proportions, necessitates a certain size of propeller, thereby limiting the revolutions.

"Friction and slip constitute the normal losses in all propellers, and augmented resistance must also be taken into account. This latter loss, however, is materially reduced with the smaller diameters of propeller found in turbine work. The percentage of slip has varied from 28 per cent. in H.M.S. 'Viper,' down to about 14 per cent. in the 'Viking'—channel steamers usually having about from 17 to 24 per cent. For large ocean-going vessels about 16 to 20 per cent. may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Comparing the 'Manxman' with the 'Ulster'—a Holyhead mail boat of similar speed and power—the total disc area of the twin propellers of the latter is 226 sq. ft. (for two twelve-foot diameter propellers), while that of the 'Manxman' is only about 80 sq. ft. If the thrust deduction is proportional to the absolute area of the disturbance of the steam lines at the stern, the effect on the 'Ulster' will be far greater than on the 'Manxman,' but this action is to some extent affected by the intensity over the disturbing area, which again is modified by the proximity of the propellers to the side of the vessel, this being less in turbine work. The disc area in H.M.S. 'Velox' is less than half that of the propellers in ordinary destroyers of the same power and speed. Cavitation is a preventable loss, and its presence on many vessels with insufficient blade area may be deduced from the falling off of the thrust curve and the rapid rise in the slip curve above a certain speed.

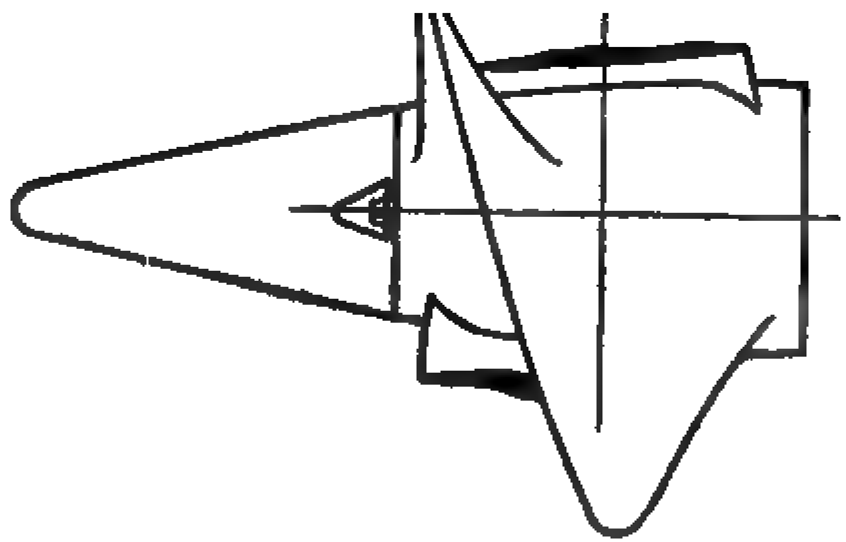
"From the analysis of numerous trials it appears that the pressure per square inch of projected area, when reduced to 12 in. immersion of tip, due to the effective thrust, is approximately 1 lb. for every 1,000 ft. per minute of circumferential velocity of blade tips. For a screw of a given pitch ratio, working at its maximum efficiency, this velocity should be proportional to the designed speed of the ship, and at full speed the pressures seem to have hitherto been about 5 lbs. per square inch for slow cargo vessels, from 6 to 7 lbs. for ocean-going mail steamers, and from 7.5 to 8.5 for cross-channel steamers; in cruisers and battleships they vary from 8 to 10.5 lbs. in some recent notable instances, in a torpedo craft from 9 to

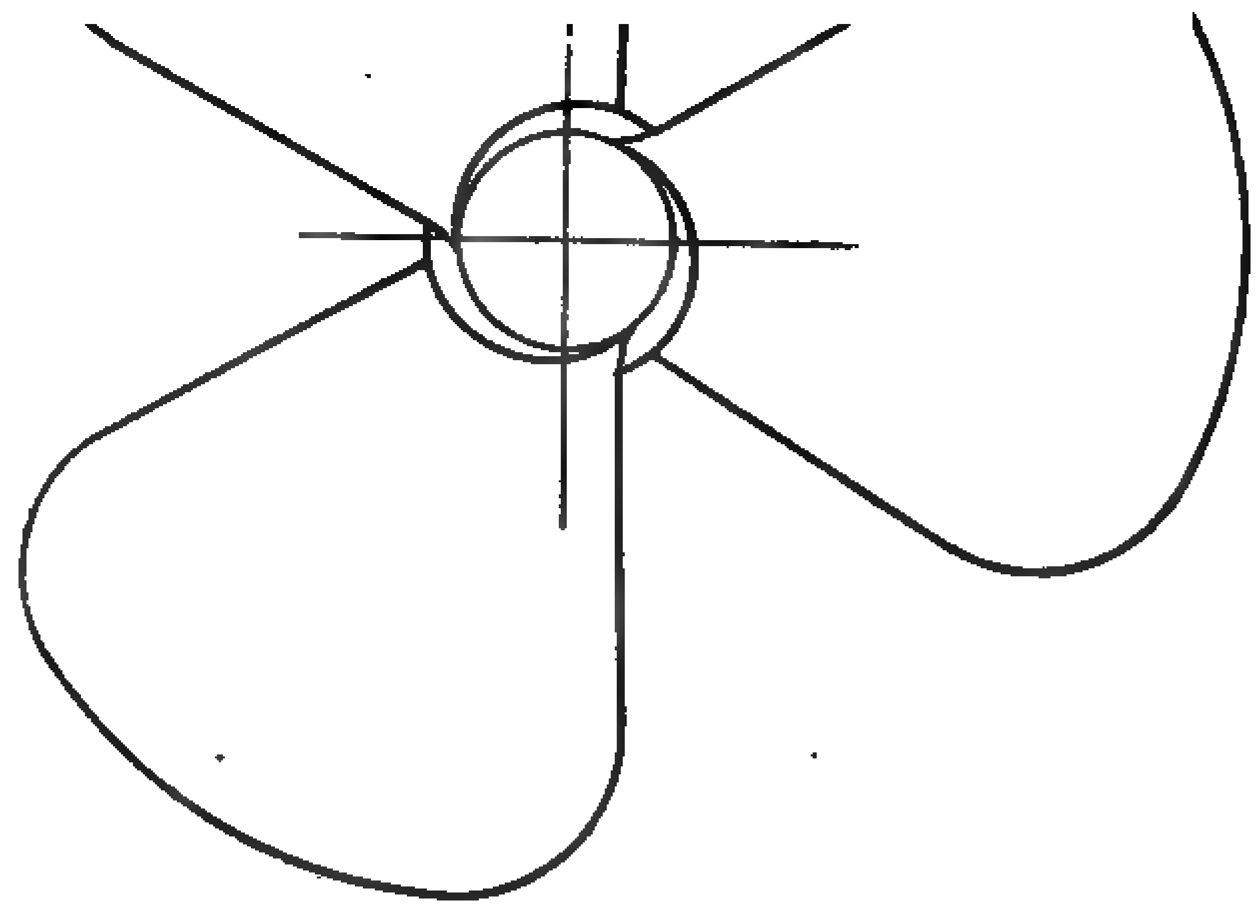


TURBINE TYPE PROPELLER

SCALE $\frac{3}{4}$" PER FOOT

PITCH .... 5′–0″
DIAMETER .... 5′–7″
EXPANDED BLADE AREA ... 11·8 SQUARE FEET
PITCH RATIO .... ·89
AREA RATIO .... ·48





BRONZE TURBINE TYPE PROPELLER

DIAMETER 9′-3″ PITCH 8′-6″

PITCH RATIO ·91 AREA RATIO ·56

SCALE ⅛ INCH PER FOOT

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PRESSURE LBS PER SQ INCH

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screws per shaft respectively driven by identical turbines, the approximate revolutions at full speed being 900, 1,200, and 1,050 for 27, 36, and 31 knots respectively on trial.

"The percentage of slip has varied from 28 per cent. in H.M.S. 'Viper' down to about 14 per cent. in the 'Viking'—channel steamers usually having about from 17 to 24 per cent. For large ocean-going vessels about 16 to 20 per cent. may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Recent experiments in the model tank at Washington, D.C.,* seem to show that a symmetrical section will materially increase the pressure at which cavitation commences, and also demonstrate that in fine-pitch high-speed screws the back of the blade should receive almost as much attention as the face. This, as the author is well aware, is no new idea, but there have been repeated indications, especially in the trials of ordinary torpedo-boat destroyers, that while the gain may not be great, it is sufficient to merit attention. Mr Parsons has advocated a 10 per cent. reduction of pitch at the blade tip in order to avoid excessive local thrust, which might induce early cavitation, but there seems to be no advantage from departing from a true screw. The tendency of late years, in reciprocating-engine practice, has been to increase the ratio of projected to disc area from the .2 of Froude's classic screw, and the .22 to .26 of naval practice, to about .33; destroyer practice is included between this and .37, or even .4, at which point turbine practice may be said to commence. In this even from .5 to .56 has been used, but beyond, about .58 blade interference becomes excessive, and to obtain greater area a larger diameter must be used.

"The best form of blade is still undetermined. In the photo of the stern of the 'Lorena' will be seen the usual shape adopted, and experience seems to show that this almost circular shape, with the area disposed symmetrically on each side of the centre line, and with the generating line of the screw at right angles to the axis, gives as good results as any form.

"I find that the following formula will give the diameter of a turbine propeller with considerable accuracy when the effective thrust along the shaft is known, and this must be calculated in any case if the steam balance of the turbine is to be good:—

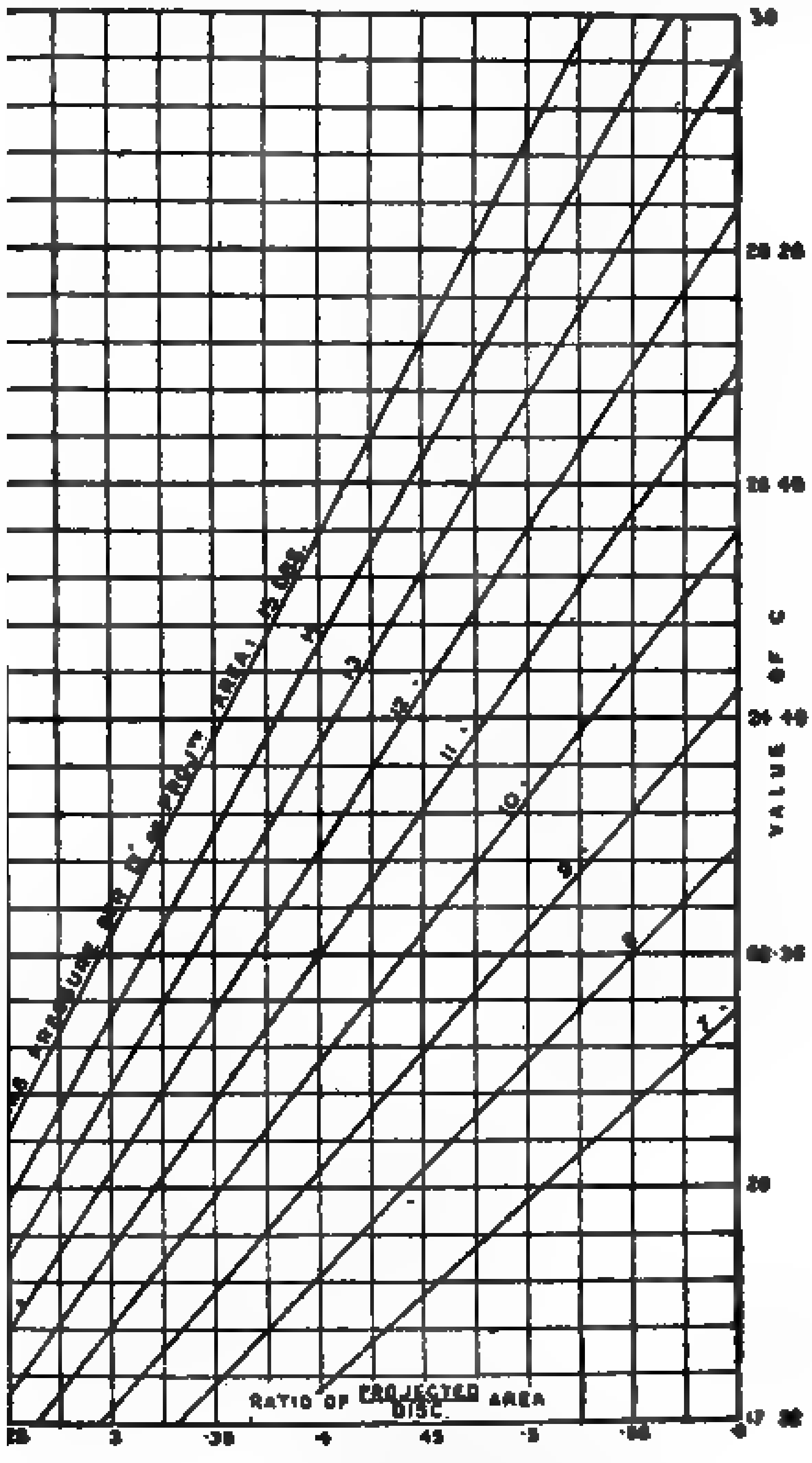
$$\text{Diameter of propeller in feet} = \sqrt{\frac{\text{effective thrust in lbs.}}{\text{coefficient}}} = \frac{\sqrt{T}}{C}.$$

"This coefficient has been deduced from the limiting pressure per square inch, and the ratio of projected to disc area, and is given in diagram form, in Fig. 2, where the full coefficient 400—900 is given in the left-hand scale, and values of C or $\sqrt{\text{Coefficient}}$ appear in the right-hand margin. The square root is only extracted for simplicity, whereby such coefficients as 30 are obtained for H.M.S. 'Viper'; while for the 'Manxman,' that for the centre screw is 26.4, and for the wing screws about 28.75. For large ocean-going vessels with lower designing pressures these values will be rather less, perhaps about 22.

"Compared with above values of C, reciprocating-engine practice gives such figures as:—R.M.S. 'Lucania,' 16.5; H.M.S. 'Diadem' class, 17.5; H.M.S. 'Exmouth' class, 19.0; and standard 30-knot destroyers of 6,100

* See D. W. Taylor's paper, American Society N. A., 1904.

which have much lower ratios of projected to disc area, and diameter and smaller C for a given power. Regarding these e understood that C is an approximation only, and, owing to



$$\text{DIA OF PROPELLER IN FEET} = \sqrt{\frac{\text{ACTUAL THRUST OF SCREW IN LBS.}}{\text{CO-EFFICIENT CORRESPONDING TO DESIRED PRESSURE AND AREA RATIO.}}} = \sqrt{\frac{T}{C}}$$

Fig. 2.—Propeller Calculations.

btaining the actual power in each shaft in every case, cannot absolutely accurate. Probably, however, the error involved ent. It seems likely, to some at present undeterminable narrow limits for each class of vessel, that the propeller

efficiency is proportional to the coefficient C, and this seems to be borne out by the trials of the 'Manxman' and the 'Londonderry.'

"Effective thrust is a somewhat subtle subject, and our knowledge of propulsive efficiency is by no means what it ought to be. These considerations will undoubtedly be brought into far greater prominence in the near future, and it is by no means improbable that the Admiralty or certain private owners will require some definite standard in this, just as coal or steam consumption is regulated at present. The more propeller efficiency is studied and understood the greater will be the improvement in the design of turbine installations for marine work; the turbine itself is a comparatively secondary consideration, and while at present propeller dimensions for turbine steamers can be quite as closely determined as those for ordinary work, the exact proportions must necessarily largely remain subject to modification from actual experience.

"The following table gives a few propeller dimensions and the corresponding coefficients, which the author trusts will be of use in designing high-speed screws:—

Propeller Dimensions.

Vessel.	Type.	No. of Screws.	Diameter.	Pitch.	Pitch Ratio.	Speed of Tip.	C Approximate.
			ft. in.	ft. in.		ft. p. min.	
Turbinia -	Experimental	9 3	1–6 2–4	2–0 2–4	1.33 1.0	10,860 ...	28 * 31.3*
Viper - -	T. B. D.	8	3–4	4–0 fwd 4–6 aft.	1.2 1.35	12,350	30.1
Amethyst -	3rd class cruiser	3_2^1	6–6	6–6 5–10	1.0 .898	9,200 10,000	30.8
Manxman -	Cross-channel steamer	3_2^1	6–2 5–7	5–7 5–0	.906 .896	10,270 10,760	26.4 28.75
Londonderry	Cross-channel steamer	3_2^1	5–0	4–6	.9	10,550 11,810	30.8
Dieppe -	Cross-channel steamer	3	5–3	...	...	10,100 10,400	29
Carmania -	Atlantic mail	3	14–0	13–0	.928	8,125	21
Victorian -	Intermediate	3	8–30	...	...	7,150	24.65

* These C values are calculated from the same effective thrust in each case.

"While a general tendency has been very noticeable towards increasing the propeller diameter and reducing the revolutions, there will, of course, be some point, at present undetermined, at which the triple screws used in turbine work will be distinctly less efficient than ordinary twin screws. Very largely this is the case at present with triple screws driven by piston engines, on account of excessive thrust deduction and interference, but probably before this point is reached the weight of the turbines will have prevented its adoption.

"Having obtained the diameter of the propeller and the revolutions possible, the design of the turbine can then be undertaken, but for this no formulæ exist at present, such as are met with in reciprocating-engine practice."

Under actual sea-going conditions, the resistance to advancement is increased by wind, &c., so that to obtain the same advance more revolutions are required, but as this increase in revolutions often produces cavitation (the propellers and revolutions being already designed for the limiting conditions), so that the limit is exceeded, and loss of push or thrust results, conjointly with a high slip ratio; hence, 15 per cent. slip may be the trial result, but 30 per cent. slip the actual sea-going result.

From the foregoing it will be evident that increasing the revolutions and decreasing the pitch may not give identical results, although theoretically this should be the case. The phenomenon of cavitation upsets the calculations, and seriously affects the results, owing to reduced thrust efficiency. Generally speaking—

1. A high turbine efficiency means a low propeller efficiency.
2. A high propeller efficiency means a low turbine efficiency.

The best combined efficiency of turbine and propeller is what has to be aimed at, and this can only be obtained by sacrificing one or other or both of the two efficiencies referred to, so that a compromise is effected. Generally, the propeller efficiency is sacrificed, as the advantage of this results in a higher proportional turbine efficiency and economy.

Turbine and Propeller Efficiency Combined.—It is often advisable in turbine steamers to sacrifice propeller efficiency so as to obtain a high turbine efficiency. This accounts for the high slip ratio noticeable in many turbine steamers, as it is found better to drop some of the propeller efficiency to gain more turbine efficiency.

Hence with high revolutions the turbine efficiency will be good, but the propeller may possibly give a better result with less revolutions per minute. A compromise is thus effected to produce the highest possible *combined* efficiency of turbines and propellers. This then explains the high slip per cent. often recorded.

Speed of Rotation.—As mentioned elsewhere, a great deal of the economy of a turbine motor depends on the high rotational velocity of the rotor, which, of course, necessitates smaller propellers, so that if the revolution speed is lowered to allow of larger propellers being used, the diameter of the turbine rotor must be increased if the same steam velocity is to be maintained: this means, in consequence, a rapid increase in weight of the rotors and casings.

From the foregoing it will be obvious that, roughly, the most advantageous conditions for the turbine constitute the most disadvantageous conditions for the propeller, and *vice versâ*, but as very elaborate and costly tank experiments have been made, and are still being carried on, the design of the propeller is being much improved, and the efficiency increased for higher revolution speeds. Experience seems to indicate that the propeller will require to meet the turbine requirements, and not the reverse.

COMPARISON OF HIGH REVOLUTION AND LOW REVOLUTION SPEEDS ON TURBINE AND PROPELLER EFFICIENCIES.

HIGH REVOLUTIONS. (Say, 500 per min.)		LOW REVOLUTIONS. (Say, 150 per min.)	
Turbine Advantages.	Propeller Disadvantages.	Turbine Disadvantages.	Propeller Advantages.
1. Smaller turbines, therefore less weight of machinery.	1. Smaller propellers, therefore less holding power, or resistance against head winds and seas.	1. Larger turbines, therefore increased weight and space required for machinery.	1. Larger propellers giving increased blade surfaces, and more holding power.
		2. Largely increased blade clearance losses, as these vary inversely as revolutions squared.	2. Less pressure per square inch on blade surfaces, with correspondingly reduced risk of cavitation.
		3. Great increase in centrifugal force due to increased weight of rotor.	3. Margin for increased speed without danger of cavitation developing.
		(The weight varies roughly as revolutions squared.)	

Then, pitch ratio $= 6 \div 6.75 = .88$,
and projected area ratio $= 17.7 \div 6^2 \times .7854 = .62$.

Horse-power per shaft $= 10000 \div 3 = 3334$, and assuming a propulsive efficiency of 51 per cent.

$$\text{Then, effective thrust pounds} = \frac{3334 \times 33000 \times \frac{51}{100}}{\dfrac{23 \times 6080}{60}} = 24072 \text{ lbs.}$$

Therefore, $24072 \div (17.7 \times 144) = 9.4$ lbs. pressure per sq. in. of projected blade area.

THE CURTIS MARINE TURBINE.

The "Curtis" type turbine is now coming rapidly to the front as an efficient marine turbine, and recent trials have brought out the fact that the consumption per brake horse-power per hour is fairly constant at both low and high speeds. This result is chiefly due to the fact that the steam, being admitted by hand-controlled nozzles, one or more of these can be shut off as required for reduced speed, thus eliminating the wire-drawing losses which occur when the main stop-valve requires to be partly closed to reduce the steam flow and shaft speed.

The Curtis turbine is of the compound impulse type, the compounding being for pressure and velocity.

Generally described, the Curtis turbine may be said to represent a combination of the De-Laval and Parsons, as nozzles similar to the De-Laval type are arranged on the periphery of the wheels through which the steam is admitted to each stage, and fixed and moving blades similar to the Parsons type are arranged within the stages themselves.

Each stage consists of a set of expanding nozzles, and two or more rows of moving blades alternating with fixed blades. Between each stage is fixed a diaphragm with expanding nozzles. The total expansion of the steam, therefore, takes place in stages in the successive expanding nozzles, the kinetic energy developed at each expansion being absorbed as it passes through the successive moving and fixed blades of each stage.

It should be noticed that, being an impulse type turbine, the steam falls considerably in pressure in the nozzles of each stage before acting on the moving blades inside the stages, and this fall of pressure results in increase of velocity, the kinetic energy thus liberated from potential energy acting solely to produce increase of steam speed.

The revolutions can thus be kept down to bring out the efficiency of the propeller, which gives this type of turbine an advantage over the Parsons type, in which propeller efficiency is often sacrificed more or less for turbine efficiency. On referring to the trial performance of the three American scouts (page 250) it will be seen that the respective revolutions of the Curtis turbined "Salem" and Parsons turbined "Chester" were, at the 22.5 knots trial, 312.5 and 473.5, giving a difference of more than 150 revolutions per minute between the two.

It may be of interest to mention that a well-known Clyde firm have taken up and are at present experimenting with the Curtis type marine turbine with the express purpose of developing it for adaptability to moderate and low speed steamers, and for combination with the Parsons type.

The following descriptions are reprinted, with kind permission, from the pages of the *Journal of the American Society of Naval Engineers*, to the President and Council of which body the author's thanks are due:—

The general construction, as well as principal details, may be understood by the description following, and by reference to plate facing page 252, representing a marine turbine of about 3,500 B.H.P. running at 475 R.P.M.

The term "stage" implies the space within two diaphragms, containing rotating and stationary vanes and a set of nozzles. The casing A, made of cast iron, is built up of several sections of semi-cylinders bolted together, the upper half being secured to the lower by longitudinal flanges. Dished heads B, in one casting, are bolted to the ends of the cylinder and contain the shaft stuffing boxes C. The packing in these stuffing boxes consists of double sectional rings of pure carbon, the space between being steam packed. The main shaft D is of wrought steel and hollow, necessarily of a large diameter to secure rigidity. The rotors E are built up of cast hubs, forged rims, and boiler-plate sides riveted and screwed. The diaphragms F consist of cast or dished plates riveted to steel rings, the inner of which has a composition boss within which the rotor hub revolves, the outer one a dovetailed extension fitting a groove in the casing, which holds each diaphragm in position. They are made in one piece, as there exists no need for removal; the upper casing, however, can be lifted up by disconnecting the flanges. The blading for each stage is made up of stationary vanes G, and rotating vanes H, the former in grooves of sectional attachments screwed to the inside of the casing, the latter in grooves of the rotor rim. The vanes are not, as in most other turbines, inserted and fastened independently or individually, but are made up in blade segments from 10 to 12 in. long, consisting of a foundation ring, the vanes and a shroud band all cast together. When the vanes are short, they may be milled out of the solid, but are usually built up as described, the vane itself being extruded from blooms of a special composition. The bodies I, containing the nozzles, are secured to the casing at each stage, and consist of a casting with apertures separated by properly formed nickel-steel plates cast in the body. The steam inlet for the ahead stage is at K and for the backing stage at L, the exhaust nozzle for both being at M. Of the nozzles those in the first stage, both ahead and backing, are expanding, all the others are parallel. The rotor shaft rests in the bearings N and revolves in oil under pressure. Steam leakage outward in the stuffing boxes (or air leakage inwards) is brought to a minimum by connecting the annular spaces, between the carbon pack-

ing rings of the gland, to some intermediary stage where the pressure is only slightly above the atmosphere.

The steam velocities through the nozzles in the various stages are so arranged as to allot one-quarter of the total energy in the steam to the first stage and one-eighth to each of the other six. This is done to obviate an undesirably high shell pressure in the first stage, in which otherwise both the bursting and distorting stress would act detrimentally on the large shell area. Losses on account of the high frictional resistance, which occurs when a fast-rotating disc revolves in dense steam, are also, thereby, materially lessened. To illustrate the working conditions of a turbine of this kind the following data are given, which are supposed to answer to a turbine developing about 1,200 horse-power. The pressures put down in the third column, which are the stage pressures, convey clearly the situation. For any other turbine of different power, but working under similar conditions, the corresponding pressures will, of course, be nearly the same.

TABLE I.

Stage.	Absolute Inlet Pressure, lbs. per square inch.	Absolute Shell Pressure, lbs. per square inch.	Throat Pressure in Nozzle, lbs. per square inch.	Square inch of Throat required, one Nozzle.	Absolute Nozzle end Pressure in lbs. per square inch.	Square inches of Nozzle end required, one Nozzle.
1	265.0	79.0	152.9	1.339	95.7	1.5048
2	79.0	41.7	45.58	4.24	45.58	4.24
3	41.7	21.2	24.0	7.8	24.0	7.8
4	21.2	10.4	12.23	14.84	12.23	14.84
5	10.4	4.9	6.0	29.25	6.0	29.25
6	4.9	2.2	2.83	60.66	2.83	60.66
7	2.2	1.0	1.27	135.0	1.27	135.0

The first-stage nozzles have each a separately operated disc valve for shutting off or opening to steam. All of the other stages are furnished with slide valves by means of which a certain number of their nozzles can be closed.

The steam flow through the turbine may accordingly be regulated to correspond with an amount of steam required for any desired horse-power, and the function of these valves thus bear the same relation to an economical variation of the power as do cruising turbines with Parsons' system. Regulating valves as described are necessary on turbines for naval ships cruising the greater part of their time at low power. For ordinary service the regulation may be accomplished by the throttle, and those valves are, therefore, limited on mercantile turbines.

Besides the independently operated stage valves pressure gauges are located on the shell to ascertain the pressure within each stage. A drain valve connects each stage in such a way as to lead the drain from a

' stage direct into the next following and successively into the avity, at which point a connection is made with the condenser.

gement.—Machinery installations with Curtis turbines for ; purposes of large power, as mentioned before, are arranged hafts in the ordinary way (see diagram facing page 248). osition of the auxiliary machinery is, in general, also similar f arrangements with reciprocating engines. Two separate ets are, however, required, one for ahead the other for back-ipes of which, with their throttle valves, connect to the nozzle the first stage. As there exists only slight end thrust on the n uneven steam balance on the vanes within the cylinder, l of the propeller thrust must be provided for in the usual n form of a thrust block, which is placed either forward or vane clearances being comparatively liberal all around, and ng an absence of dummy pistons with their close settings, nicrometer adjustments in the thrust block entirely unneces-auges are, however, provided at the front and back of the o enable the precise location of the rotor, with reference to nary parts, being determined.

;n.—The real difficulty when starting out in designing a bine centres around the ability to determine the efficiency, or, vords, to closely approximate the pounds of steam needed per ver for given conditions of pressure at entrance and exhaust ed stipulations as to the quality of the steam. To do this in ne, and especially in turbines of the impulse type, like Curtis val, it is necessary to base one's assumptions on data secured r by actual experiments. Such tests have been performed fully with the present seven-stage Curtis marine type. An esult of such tests justifies us to assume a steam consumption . per brake horse-power, using dry saturated steam at an ssure of 265 lbs. absolute and a vacuum of 28 in., the rotor d at the same time to be approximately between 180 and r second and the allotment of power as previously stated e seven stages.

the use of superheated steam, for which this type of turbine ilarly well adapted, the consumption will undoubtedly be bly lowered, down to 13 lbs. or better, which has been amply various stationary turbine plants.

otal available heat per pound in steam expanding adiabati-ween 265 lbs. and 1 lb. absolute, determined by the formula

$$H_1 - H_2 = Q_1 - Q_2 + H_v - T_2 (E_{\theta 1} + E_v - E_{\theta 2})$$

$$2 - 70.0 + 825.6 - 563.3 (.5728 + .952 - .1329) = 352.2 \text{ B.T.U.}$$

of the heat contained in the steam could be converted into the shaft the consumption of the ideal turbine would be

$$\frac{)}{8} = 7.21 \text{ lbs. per horse-power hour.}$$

The actual turbine, on the basis of 14.0 lbs. steam per horse-power hour, will accordingly give an efficiency $= \frac{7.21}{14.0} = 51.52$ per cent.

The losses in an impulse turbine may be attributed to the following causes:—

1. Steam friction in nozzles.
2. Steam friction in vanes.
3. Steam shock on the vanes caused by deviation in the relative entrance velocities and by starting the quiescent steam between the vanes when passing nozzles.
4. Resistance to the revolving surfaces in steam of more or less density.
5. Radiation of heat from the cylinder, spreading and leakage of steam between vanes, nozzles, and shaft bushings.
6. Loss of energy due to the exit velocity.
7. Mechanical losses, such as friction of journals and stuffing boxes.

The Nozzle.—Weight, space, and propeller speed renders it necessary, in marine turbine installations, to limit the rotor diameter as well as its speed of revolutions, and incidentally to use low steam speeds. These are obtained by arranging the turbine for a comparatively large number of "expansions" or stages and by using appropriate types of nozzles.

The general principles of steam expansion and the velocity attained by the jet in different nozzles has been written up extensively in various treatises and does not come within the scope of this paper. Suffice it to say, therefore, that the jet velocity depends essentially upon the "ratio of expansion," or on the proportions observed in the areas of throat and outlet end of the nozzle and upon the initial and terminal pressure at the orifice and outlet; moreover, that a definite relation always exists between the initial pressure and that of the throat, when the orifice is well rounded. Thus if P signifies the initial pressure, the throat pressure will be .577 P if the outlet pressure is equal to this or any other pressure below .577 P down to a perfect vacuum. This occurs in either a parallel or an expanding nozzle.

Action.—The Curtis marine turbine is of the compound impulse type. The steam expands in seven sets of nozzles or pressure stages successively from the initial pressure to that of the exhaust. The first pressure stage, as has been mentioned, develops one-fourth the total energy, the other six each one-eighth. The jet energy is transformed into work by impulse on the moving discs, which, according to the jet velocity used, are arranged in from three to four velocity stages. Good efficiency demands that certain proportions of steam and bucket velocity be observed throughout the turbine.

The steam action is essentially as follows: After expansion in the nozzles of the first stage the steam issues in solid jets against the first row of moving buckets, which absorb a part of the jet energy,

passing through said buckets, meets the first row of
/anes. The purpose of these vanes is to guide the steam
:ond row of moving buckets, which, in their turn, take up
tion of the kinetic energy still possessed by the fast flowing
e same time diverting it into the second row of stationary
ich deflect the steam on the third row of moving buckets,
n more of the energy is taken up. In the first stage,
nitial nozzle velocity is considerably higher than in any

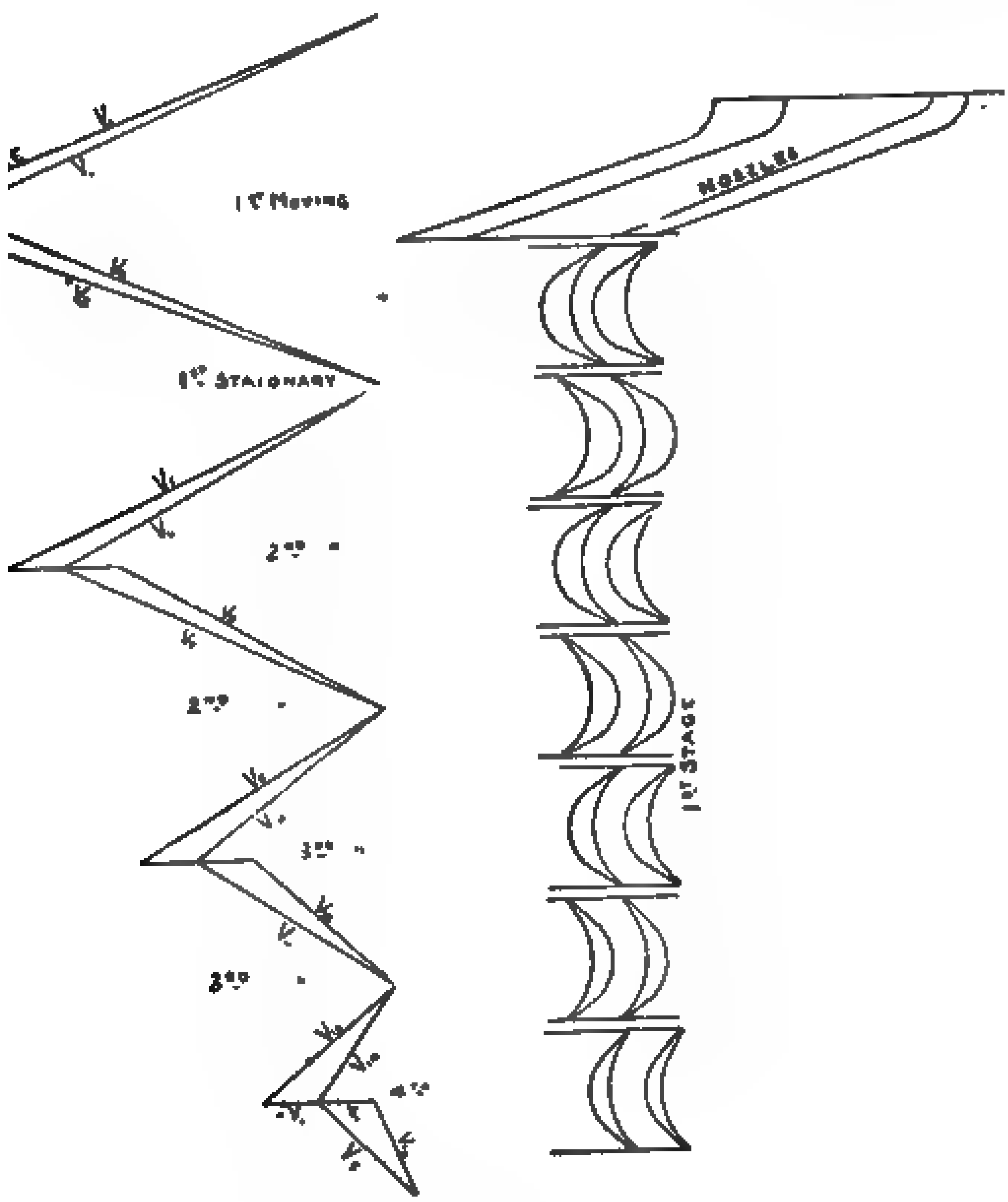


Fig. 5.
Velocity Diagram, "Curtis Turbine," first stage.

stage, this operation is again repeated in a third stationary
fourth moving row, the remaining energy after that being
ɔ warrant additional buckets. The pressure at the outlet
ıozzles is brought down to the pressure within each stage
pansion in the various rows of buckets, the volume of the
sponding to successive pressures. However, due to the
e velocity of the steam is gradually diminished by the
ıbsorption of energy, the passages traversed by the steam

must be enlarged. This is provided for by lengthening the buckets as well as by increasing the vane angles in each succeeding row.

After leaving the last row of moving buckets in each stage the steam attains partial rest before it enters the nozzles of the next stage, in a manner similar to that which occurs between each "expansion" of a Parsons turbine. On entering the nozzles of the second stage the steam again expands, whereby new velocity is given, and now acts in the various rows of buckets of that stage exactly as it did in the first stage, and so on right through all of the seven stages of the turbine. There is this difference, however, that, owing to the nozzle velocity being very much less in the stages succeeding the first, three moving and two stationary rows will suffice there instead of respectively four and three of the first stage. The number and size of the nozzles in the different stages must obviously conform to the velocity and the volume of the steam as a result of expansion through the turbine. Due to this fact we find the nozzles circumscribing only a small arc in the first stage, gradually increasing in the following, until, in the last stage, the entire circle is completely filled with nozzles. This latter condition, however, is governed wholly by the power in comparison with the rotor diameter.

Steam velocities through the buckets of the first stage relatively to a fixed vane speed are shown diagrammatically in Fig. 5. No account has been taken of velocity increase as a result of expansion in the buckets in this diagram, which must be done in figuring vane dimensions.

Calculation.—In a steam turbine, the potential energy of the admitted steam is converted into kinetic energy of the steam jet and the kinetic energy of the steam jet into impulse force on the revolving wheel. This double transformation can take place either simultaneously in the same wheel (reaction turbine), or successively in stationary nozzles and revolving wheels (action or impulse turbine).

In the reaction turbine, steam enters the revolving wheel at admission pressure at moderate velocity, expands in the wheel and leaves the wheel at the velocity of expansion relatively to the wheel, that is, in a vane completely reversing the jet, at an absolute velocity equal to expansion velocity minus wheel velocity. To take the full energy out of the steam, it must leave the wheel at zero absolute velocity; that is, the wheel velocity must equal the velocity of the steam jet.

In the impulse turbine, the steam expands in a separate nozzle, strikes the rotating wheel at expansion pressure and expansion velocity, and rebounds therefrom, as an elastic body, with the same velocity, minus the friction loss, as that with which it strikes, hence losing velocity by an amount equal to twice the wheel velocity. To take the full energy out of the steam, the wheel velocity must therefore equal one-half the velocity of the steam jet.

In an impulse turbine, therefore, such as the Curtis turbine, with a given pressure range per wheel, the wheel velocity at best efficiency is only about 50 per cent. of what it is in a reaction turbine, and $\frac{1}{\sqrt{2}}=70$ per cent. of what it is in a combination turbine alternating between impulse and reaction effect, such as the Parsons turbine.

Since the most difficult condition of turbine design is the high rotor velocity required to utilise the high steam velocity, the impulse turbine is theoretically superior. It is not possible to get efficiency with a single turbine wheel of the reaction type, while with a single wheel of the impulse type good efficiencies have been reached by using extremely high peripheral speeds (De-Laval turbine). In general, however, while hydraulic turbines are always designed with a single wheel, in the steam turbine it is not possible to make effective use of the total energy of the steam by a single wheel, but a number of wheels in series have to be used. That is, the rotor velocity is reduced by subdividing the total pressure range into a number of successive stages. In the combination impulse and reaction turbine of Parsons, each expansion or pressure stage requires a revolving wheel and a stationary guide wheel, the steam acting by impulse at entrance and by reaction at the leaving edge.

In the impulse turbine, in which the expansion of the steam is carried out in nozzles separately from the vane system, a further effective means of speed reduction is presented by the use of several velocity steps in each stage; that is, by imparting the velocity of the current of steam to a number of successive wheels composing a single revolving disc, with stationary intermediate guide wheels.

Since in a revolving wheel of an impulse turbine the steam velocity is reduced by an amount equal to twice the wheel velocity for maximum efficiency, a single wheel per stage must revolve at one-half the steam velocity, two wheels per stage at one-quarter steam velocity, &c. The use of several wheels per stage, therefore, is a more effective means of reducing the rotor speed—or inversely, at given rotor speed, reducing the total number of revolving wheels—than the use of several expansion steps. A two-wheel stage can take care of twice the steam velocity (that is, four times the steam energy) of a single-wheel stage, and therefore replaces four single-wheel stages; or, in other words, the speed reduction of the rotor is proportional to the number of wheels per stage, that is, the number of velocity steps; on the other hand, it is proportional to the square root of the number of stages, that is, number of pressure steps.

The simultaneous use of pressure steps, or expansion stages, and velocity steps, or number of wheels per stage, therefore, leads to a construction requiring a comparatively small total number of wheels, as carried out in the Curtis type of turbine.

The following items are especially interesting with respect to the impulse type of turbine in review with the compound reaction type:—

1. For a given bucket speed and pressure range the number of vane rows is considerably less in the former than in the latter.

2. The steam expansion being executed in nozzles renders possible the use of high initial pressures, as well as superheat, without affecting detrimentally the turbine cylinder from excessive stress, or the vane system from distortion incidental to superheat.

3. Aside from throttle-valve regulation the steam supply may be varied, to suit any degree of power to be developed by the turbine, by shut-off valves on the nozzles, eliminating thereby the necessity for separate cruising turbines as now usually arranged for in warships.

4. Large clearances are used in Curtis marine turbine at both side and end of blades, thus minimising the danger of fouling of buckets from displacement of shaft or vibratory influences.

5. Balance or dummy pistons, to equalise end pressure, do not exist in the impulse marine turbine, and a source of loss, as well as the necessity for delicate adjustment, is eliminated.

6. For the same total power propeller diameters are larger in twin-screw ships fitted with Curtis turbines than in vessels of triple or quadruple screws with Parsons. The capacity for manœuvring is increased by using larger propellers and by making available all of the blade area against the two-thirds to one-half provided for in turbine ships of latter screw arrangement. Backing turbines are, however, fitted on each shaft in quadruple screw arrangements of naval ships.

Appended will be found a description of the Curtis turbines at present under construction at the Fore River Shipbuilding Company for the Japanese cruiser "Ibuki" of 14,600 tons displacement:—

Curtis Turbines of Japanese Cruiser "Ibuki."

The motive power of the "Ibuki" consists of two turbines, which are designed to develop a normal horse-power of 24,000, sufficient to drive the "Ibuki" at 22 knots speed. They are intended, however, to develop a maximum overload horse-power of 27,000, which should suffice to give a speed of nearly 23 knots. The rotor is 144 in. in diameter. The casing has an outside diameter of 14 ft., and a length over all of 17 ft. The weight of the two turbines together is 360 tons.

Each turbine consists of a cast-iron cylindrical casing divided by dished diaphragms into a series of separate compartments. In each compartment or "stage" there is a separate wheel, which carries on its periphery three rows of moving buckets (for reasons later described the first wheel has four rows). The wheels are all mounted on a hollow steel shaft carried by two bearings. Where the shaft passes through the diaphragms they are provided with bronze bushings having a small clearance, thus preventing appreciable steam leakage from one stage to the other. Where the shaft passes out through the ends of the casing it is provided with carbon stuffing boxes, which prevent steam leaking out at the ahead end, or air leaking in at the back end where a vacuum exists.

The stuffing boxes are supplied with steam in the space between the carbon packing to prevent air leaking in and lowering the vacuum. They are also drained to the fourth stage shell.

iteel steam chests for ahead and astern running are attached ont and back casing heads, and are flanged for the main ›es. The nozzles for each stage are bolted to the diaphragms, ragms having steam-port openings cast in them to allow the pass through to the nozzles.

:uvring is accomplished by means of two lever-operated throttle valves, each taking steam from the main steam : delivering to the ahead steam chest and the other to n steam chest. There are seven ahead wheels and two heels. The reverse wheels are mounted in the after end of z, and under ordinary ahead running they are in a vacuum efore, do not waste power by steam friction. They are the ahead wheels, except that the blades are reversed. To hen going ahead, the ahead-throttle valve is shut and the ırottle valve opened, which is easily and quickly accomplished ›erating levers of the two throttle valves.

pipes are provided, connecting each stage with the next, so condensed steam in any stage will pass to the next one of ssure, and there give up a part of its heat to do useful work. aust chamber drains to the condenser and the discharge l by a small steam ejector. A regular marine thrust bearing :d to the forward end of the turbine shaft. In addition the propeller thrust, this bearing also maintains the proper :tion of the rotor, so that the axial clearance of the blades This clearance is one-tenth of an inch on the first wheel ases to one-quarter of an inch on the seventh wheel. The put at the forward end, so that any unequal expansion of and casing will be allowed for at the aft end, where the is largest. This axial clearance is very ample to allow for ıal heat expansion that may occur and any mechanical ties, and leave sufficient leeway for adjustment.

low for the increased volume of the steam as it expands g from stage to stage at lowering pressures, the lengths ades are increased and also the arc of the nozzles is in- hus giving greater area of passage in each succeeding stage. any one stage, the blade lengths are increased in each g row, because the velocity falls as the steam passes from w, although it is at practically constant pressure throughout

ler to keep the pressure in the shell as low as possible, the listribution is arranged so that one-fourth of the available the steam is expended in the first stage and one-eighth the other stages. This requires the first-stage nozzles to be panding type, but all the other nozzles are of the parallel- . Also, the first-stage wheel is provided with four rows

Shop Test of 12,000 H.P. Curtis Turbine of Japanese Cruiser "Ibuki."

Constructed by Fore River Shipbuilding Co., Mass., U.S.A.

"The Marine Steam Turbine." [*To face page* 240.

of buckets instead of three, as on the other wheels, since the greater energy drop produces greater velocity of the steam jet from the nozzles, which requires more rows of buckets to properly absorb the energy at the bucket speed used. This arrangement makes all the ahead wheels except the first operate under eight-stage conditions. The principal advantages of the Curtis design of marine steam turbine are as follows: Small number of blades; large clearance around blades; strong mechanical construction of blading; economy at reduced speed, without cruising turbines; interior of shell not subjected to full steam pressure; low revolutions for given horse-power; absence of dummy pistons and packing.

The small number, large clearance and strong construction of the blades make blade stripping practically impossible, and no case has occurred.

By the use of valves on the nozzle openings of the diaphragms, the proper steam-pressure distribution can be maintained at reduced steam flow, thus keeping up the economy at low speed of vessel, except, of course, for the unavoidable loss due to lower revolutions and dispensing with cruising turbines.

Full steam pressure comes on the steam chest only, which is a comparatively small steel casting. The greatest pressure in the turbine shell is less than one-third the working steam pressure. This permits high steam pressure to be used, and large turbine diameter in comparison to the power. It also reduces expansion difficulties.

The comparatively low revolutions permissible for a given power without sacrifice of economy or excessive weight allows the twin-screw arrangement to be used instead of three or four screws. Also other conditions being the same, lower revolutions will give a higher efficiency of the propeller. Low revolutions also permit the use of turbines in comparatively (for turbine vessels) low-speed vessels.

Absence of dummy pistons and their packing eliminates the leakage of high-pressure steam and makes the economy independent of any adjustments, so that the initial economy will be maintained continuously and will not be affected by any wear. — *Scientific American.*

Trials of U.S. Scout "Salem."

The following notes, referring to the trials of the U.S. cruiser scout "Salem" and of the Curtis type turbine, are reprinted from a booklet issued by the Fore River Shipbuilding Company, Quincy, Massachusetts, U.S.A., with the kind permission of the firm mentioned

"The change from kinetic energy to work is achieved by 'impulse' due to jets of steam acting upon blades formec 'wheels' mounted on the shaft to be rotated. The steam exp in a number of sets of nozzles or pressure 'stages' successively the high pressure to the exhaust end of the turbine. Thus, expanding in the nozzles of the first 'stage,' the steam issues in against the first row of buckets on the rotating wheel, a large pa

the energy being absorbed. It then flows to a row of stationary vanes, which guide the steam into a second row of moving buckets. These may be followed by a second set of fixed vanes and a third set of moving ones, after which the steam leaves the 'stage,' as it is called, through a second set of nozzles, where further expansion takes place, again generating velocity. From these nozzles it flows once more in sinuous fashion through successive sets of moving and fixed blades, and thence to other 'stages.' The important point to note is that expansion of the steam takes place only in the nozzle, and not in either the fixed or moving blades. Hence the pressure of the steam does not alter between one set of nozzles and the next. At the low-pressure end the nozzles cover the whole periphery of the wheel, but at the high-pressure end they extend only over an arc often not more than one-eighth of the whole circumference. It is thus possible to reduce the power of the turbine by cutting out a proportion of the total number of nozzles, instead of by reducing the pressure of the steam supplied by throttling it at the valve. Thus, whereas in the Parsons system cruising turbines are fitted to attain reasonable economy at low speeds, they are unnecessary with the Curtis system.

"The Curtis turbine occupies a middle position between the high-speed Parsons and the low-speed reciprocating engine; and, because of the moderate speed of revolution, and the fact that the power can be developed upon two instead of four shafts, it has become possible to secure a high propeller efficiency."

. . . .

"The propulsive efficiency of the 'Salem' rose from 55 per cent. at 12 knots to a maximum of 62.8 per cent. at the contract speed of 24 knots, then fell, with the increase of slip, to 62.4 per cent. at 25 knots and 59.4 per cent. at 26 knots. This is a remarkable result for a turbine equipment, and comes pretty near to the efficiency of the crack German liners, which have shown as high as 67 and 68 per cent. The present propellers were adopted after a series of trial runs with four different designs of propellers: one by the Navy Department; another by the Denny firm, Scotland; a third by the Vulcan Works, Germany; and the fourth by the Fore River Company. The Government design broke down through excessive cavitation early in the trials. The Denny propellers showed 50 per cent. efficiency at 24 knots, the Vulcan 54.04 per cent. at 24 knots, and the Fore River type, which was designed by the chief engineer, Mr Charles B. Edwards, showed 62.7 per cent. at 24.5 knots. We present two illustrations of these propellers, which are 9 ft. 6 in. diameter, with a pitch of 8 ft. 8 in., that will possess strong interest in connection with these comparative figures.

"The standardisation trials held to determine the number of revolutions of the propellers corresponding to various speeds, from 12 knots to the highest speeds of the vessel, took place off Rockland in from 40 to 60 fathoms of water. The start and end of the mile are marked by pairs of posts set up on shore, and the time is taken from the

Lowering Rotor into Casing of 12,000 H.P. Curtis Turbine.

Constructed by Fore River Shipbuilding Co., Mass., U.S.A.

[To face page 242.

small clearance at the ends of the blades, as there is no tendency for the steam to leak around the ends. A large clearance can thus be used, greatly improving the mechanical reliability.

"(4) With the Curtis design one turbine only on each shaft is necessary for economical operation at all speeds; and equal results can be obtained as with a Parsons installation using two extra cruising
one high and one intermediate pressure.
ʼull steam pressure only comes on the steam chest, which is
ıtively small steel casting, and the greatest pressure in the
ıell is less than one-third the full working steam pressure.
nits the use of a high working pressure and large turbine
, and greatly reduces the chance of any difficulty from heat
ı. It also permits the turbine to be started very quickly
:old condition without danger of unequal expansions causing
'hile the Parsons turbine requires gradual warming up and
ıtching of the clearance during the process.
\bsence of dummy pistons and their packing eliminates the
f high-pressure steam and makes the economy independent
ljustments, so that the initial economy will be maintained
.sly. It also eliminates the necessity of maintaining the very
adjustment required for dummy-piston packing strips, and
r of their fouling.
Jnder given conditions the Curtis will run at lower revolu-
s permitting twin screws to be used. This gives a higher
: efficiency and makes the entire propeller area available
ıg. It also greatly simplifies the entire machinery installa-
ecially controlling valve arrangement, which gives much
nœuvring qualities.
)ue to the simple controlling-valve arrangement used with
vs, the large clearances around blades, the absence of dummy-
:king, the substantial blade construction, and the low steam
n shell, as above described, it is possible to throw the turbine
from full ahead to full astern with entire safety. In the
esign reversing must be done with considerable care to avoid

\s previously described the Curtis turbine can obtain speed
r using either the throttle or steam-chest nozzle valves, or
ıtion of both, while the Parsons turbine only has the throttle.
win-screw installations, as used by Curtis, each turbine is
:ntly controlled with great certainty, while in Parsons installa-
ı one turbine exhausting into another, each screw cannot be
:ntly controlled, and actual speed of the vessel cannot be so
r known.
As above described the Curtis turbine can handle without
ırge doses of water in the steam supply, which would strip
.g of the Parsons design."

. . . .

Principle of Operation.—"The Curtis turbine is of the impulse type, and, in order to reduce the economical speed of rotation to a point suitable for direct driving of marine propellers, the turbine is divided into several pressure stages, and each pressure stage is provided with several rows of revolving buckets. The simplest possible impulse turbine would consist of a single wheel having a single row of buckets on which a jet of steam is directed by a suitable nozzle. Examples of such a turbine are the De-Laval steam turbine and the Pelton water wheel.

"A simple single-wheel impulse turbine has its maximum economy when the buckets move with a velocity of approximately half that of the jet of steam. A jet of high-pressure steam, in a proper nozzle, will attain a velocity of about 4,000 ft. per second when discharging into a vacuum, which would require the enormous speed of 2,000 ft. per second for the buckets to obtain the maximum economy. This speed would require such a high number of revolutions that it would

"Stage" Diagram of Curtis Type Turbine.

render direct connection to a propeller impossible, and would also produce centrifugal forces too great to be properly handled.

"By using several wheels on the same shaft, each one utilising the exhaust from the one preceding, the pressure drop and corresponding steam-jet velocity to be handled by each wheel is greatly lessened. Thus by using eight wheels, each in a separate chamber with its own nozzle, and so proportioning the steam pressure that the available energy is equally divided, the steam-jet velocity is reduced to about 1,400 ft. per second and the bucket speed to 700 ft. per second. This, however, is still too high for satisfactory marine work, and, in order to lower it, each wheel is provided with three rows of moving buckets instead of only one. Each row of moving buckets will take out from the jet velocity approximately twice its own speed, so that the use of three rows on each of eight wheels will give a bucket speed of 230 ft. per second. This speed can be used satisfactorily on high-speed vessels, but as the economy curve is quite flat at the point of

maximum efficiency, the speed can be somewhat reduced without greatly reducing the economy, giving a fair working speed of about 160 ft. per second. The above-described method of reducing the economical bucket speed by means of several separate stages, each having several rows of buckets, is illustrated in diagrammatic form which shows the first two stages of a turbine of which the remaining stages would be similar. It is seen that between each row of moving blades there is a row of stationary blades, the object of which is to reverse the direction of flow of the steam as it leaves one row of moving blades and direct it into the next row of moving blades. The arrows show the direction of the steam as it passes through the nozzle and then through the moving and stationary blades.

"When the steam passes through a nozzle its pressure drops and it gains velocity; this velocity is absorbed by the moving buckets, thus converting energy of the steam into mechanical work. As the steam passes from stage to stage it drops in pressure, and when it passes from one row of buckets to the next in the same stage it loses velocity, but the pressure is the same in any one stage in all the bucket rows, and is also the same on both sides of the wheel.

"This uniform pressure in all parts of stage obviates the necessity of small radial clearance between the ends of the blades and the casing, since there is no leakage around them. This clearance is, therefore, made as large as desired for proper mechanical construction.

"The axial clearance, or distance between the edges of the blades, is also made quite large, its only limit being the slight disturbance of the jet which occurs at some distance from a nozzle or guide blade."

Description of Construction of "Salem's" Turbines.—The plate opposite is a vertical section of one of the turbines as fitted to the "Salem," and clearly shows the construction. The turbine consists of a cast-iron cylindrical casing divided by dished cast-iron diaphragms into a series of separate compartments (cast steel is used for two first stages, for strength). In each compartment or "stage" there is a separate wheel which carries on its periphery three rows of moving buckets (for reasons later described the first wheel has four rows). The wheels are all mounted on a hollow steel shaft carried by self-aligning bearings as shown. Where the shaft passes through the diaphragms they are provided with bronze bushings having a small clearance, thus preventing appreciable steam leak from one stage to the other. Where the shaft passes out through the ends of the casing it is provided with carbon stuffing boxes, which prevent steam leaking out at the ahead end or air leaking in at the back end where a vacuum exists.

The stuffing boxes are supplied with steam in the space between the carbon packing to prevent air leaking in and lowering the vacuum. They are also drained to the fourth-stage shell.

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Cast-steel steam chests for ahead and astern running are attached to the front and back casing heads as shown, and are flanged for the main steam pipes. The nozzles for each stage are bolted to the diaphragms as shown, the diaphragms having steam-port openings cast in them to allow the steam to pass through to the nozzles.

Manœuvring is accomplished by means of two lever-operated balanced throttle valves, each taking steam from the main steam pipe, and one delivering to the ahead steam chest and the other to the astern steam chest. There are seven ahead wheels and two reverse wheels. The reverse wheels are mounted in the after end of the casing, and under ordinary ahead running they are in a vacuum and therefore do not waste power by steam friction. They are the same as the ahead wheels, except the blades are reversed. To reverse

A B C D

Blading Segment of Curtis Turbine.

A, Blades held in core previous to casting on Base and Shroud.
B, Blade cut to length and notched.
C, Segment after casting on base and shroud.
D, Segment machined to size.

when going ahead the ahead throttle valve is shut and the reverse throttle valve opened, which is easily and quickly accomplished by the operating levers of the two throttle valves.

In the ahead steam chest there are twenty valves, each opening one of the nozzles for the first-stage wheel. For continuous running sufficient of the nozzles are opened to give the speed desired, and the ahead throttle valve is left open, thus giving full pressure in the steam chest. Sixteen nozzles will give full designed power, leaving four for overload. The astern steam chest has the same number of nozzles, but only eight have valves. For manœuvring, the nozzle valves in the steam chests are left open and the speed is controlled by the throttle valves.

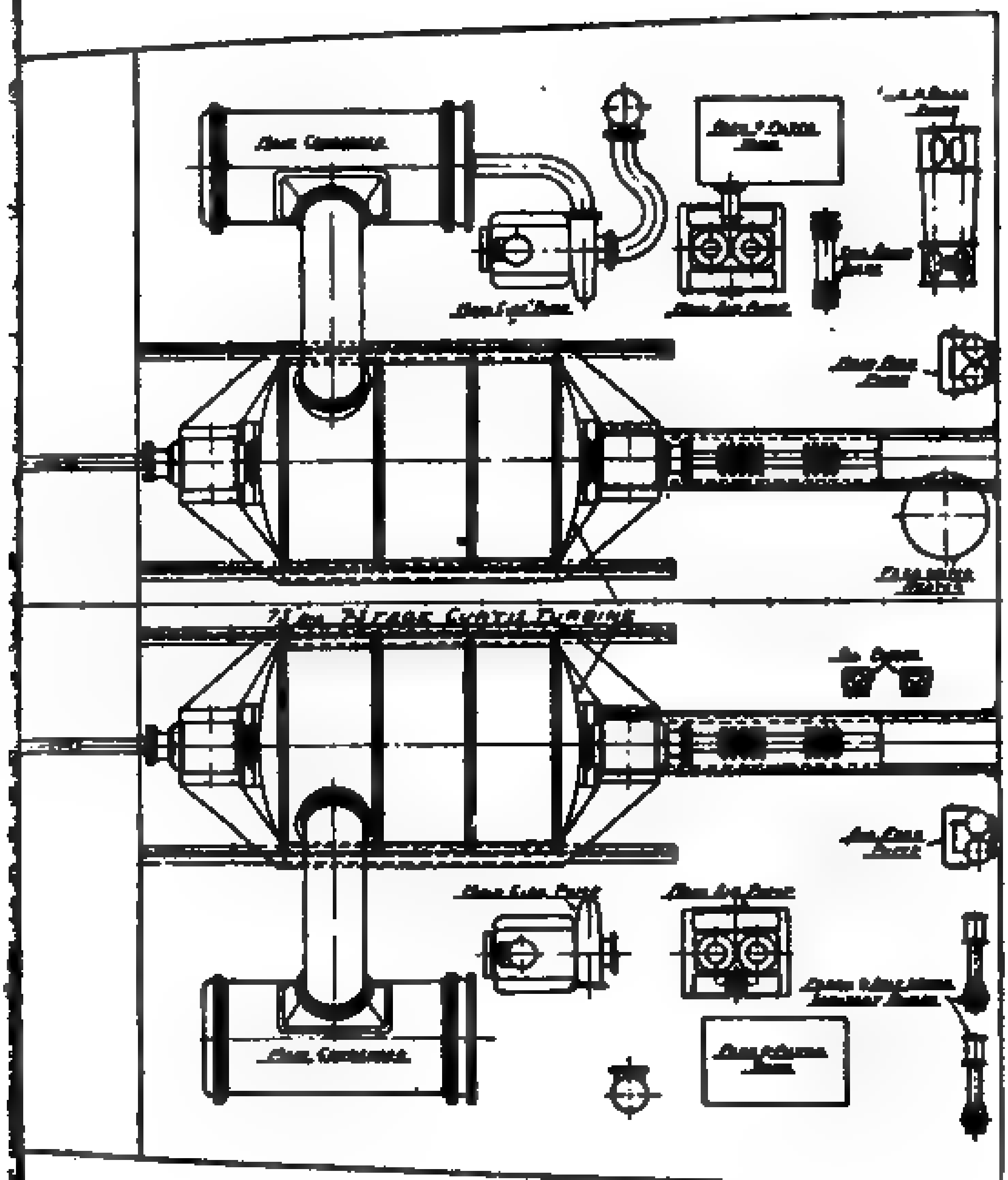
Drain pipes are provided connecting each stage with the next, so in any stage will pass to the next one of lower

pressure and there give up a part of its heat to do useful work. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. A regular marine thrust bearing is attached to the forward end of the turbine shaft. In addition to taking the propeller thrust, this bearing also maintains the proper axial position of the rotor, so that the axial clearance of the blades is correct. This clearance is $\frac{1}{16}$ of an inch on the first wheel and increases to $\frac{1}{4}$ of an inch on the seventh wheel. This axial clearance is very ample to allow for all unequal heat expansion that may occur and any mechanical irregularities, and leave sufficient leeway for adjustment.

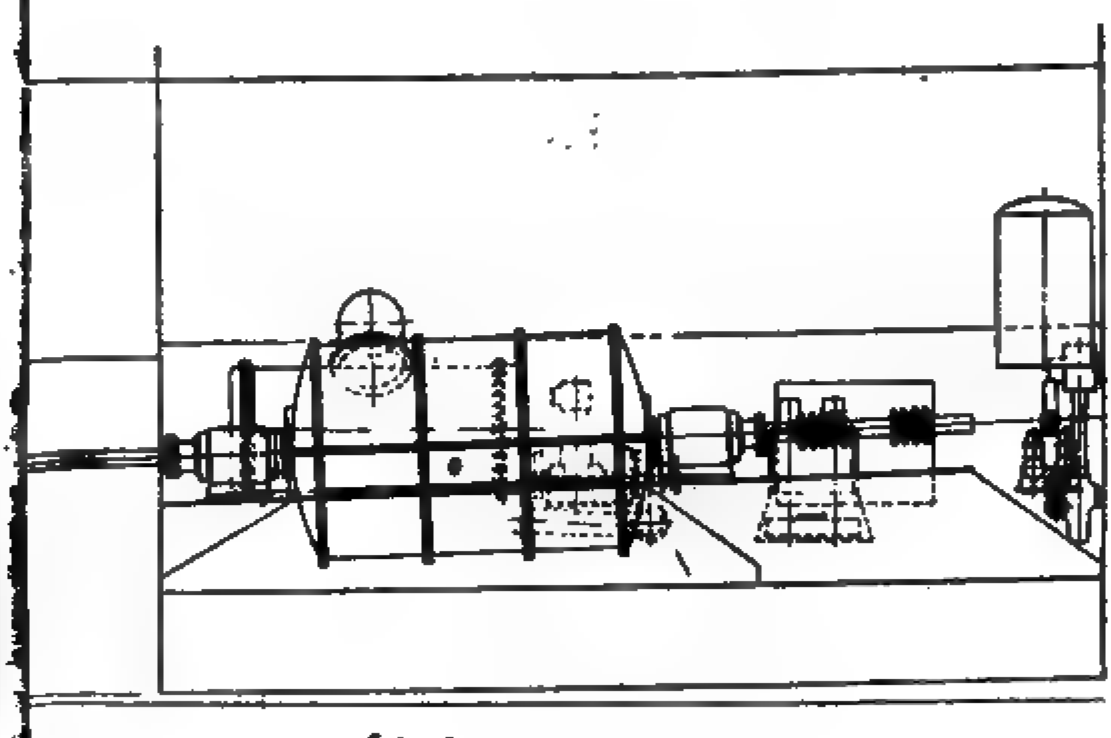
The construction of the blading is shown in the illustrations. The blades are made of special bronze accurately formed to the required

Latest Type Curtis Turbine Blading.
Consisting of Steel Channel Base with saw cuts to allow of flexibility in fitting, Steel Plate Shroud, and extruded Metal Blades.

shape by being extruded through a die and thus made into long bars. Th are cut into lengths as required by each stage, and each length :hed or drilled through at the ends, as shown at B. A number des are built up into a segment by casting on a composition n the inner ends and a shroud on the outer ends as shown at C. er to do this casting the blades are held in a sand core, with the rojecting slightly, as shown at A. After casting, the base and l are milled off true, thus completing the segment, as shown at D. e ends of the blades fuse into the cast parts, thus making cally one solid piece of the entire segment. If any blade should ippen to fuse, the metal will flow into the notch or hole and ally keep it in place. The extruding process gives the blades l, smooth skin, which reduces steam friction and effectually ts wear and erosion of the blades.



PLAN



ELEVATION

[*To face page* 248.

The segments are held in the wheel rims by inserting the bases in rectangular grooves around the steel rims and caulking the edges of the grooves. This caulking is done by means of a pneumatic caulking tool mounted on special adjustable pedestal, while the wheel is slowly revolved.

The bases and shrouds are overhung slightly beyond the edges of the buckets, so that if by any accident the wheels should be moved axially the blades would not come in contact and be damaged, as the overhanging bases and shrouds would protect them. These bases and shrouds are very strong, and can stand forcing together so hard that the turbine would be brought to a stop without any serious damage occurring. This very important feature renders stripping of the blades a practical impossibility in this design of turbine.

As before stated, the radial clearance at the ends of the blades can be made as large as desired without having any effect on the operation, and, as constructed, it varies from $\frac{1}{4}$ in. to 2 in., depending upon the proximity of the stationary blade holders, as is clearly shown in the illustration.

To allow for the increased volume of the steam as it expands in passing from stage to stage at lowering pressures, the length of the blades is increased, as shown in sectional view, and also the arc of the nozzles is increased, thus giving greater area of passage in each succeeding stage. Also, in any *one* stage, the blade lengths are increased in each succeeding row, because the velocity falls as the steam passes from row to row, although it is at practically constant pressure throughout the stage.

In order to keep the pressure in the shell as low as possible, the pressure distribution is arranged so that one-fourth of the available energy of the steam is expended in the first stage, and one-eighth in each of the other stages. This requires the first-stage nozzle to be of the expanding type, but all the other nozzles are of the parallel-flow type. Also the first-stage wheel is provided with four rows of buckets instead of three as on the other wheels, since the greater energy drop produces greater velocity of the steam jet from the nozzles, which requires more rows of buckets to properly absorb the energy at the bucket speed used. This arrangement makes all the ahead wheels except the first operate under eight-stage conditions.

Reciprocating and Turbine Engines compared.—The results shown below of the first series of competitive trials between three American vessels of the "cruiser scout" class afford an interesting comparison of performances, and prove conclusively the superiority almost throughout of the turbine over the ordinary reciprocating engine.

The "Birmingham" is fitted with reciprocating engines, the "Salem" with Curtis type turbines, and the "Chester" with Parsons type turbines, otherwise the three steamers are identical in dimensions and displacement. The value of the results is greatly discounted by the fact that the shaft horse-power of the "Chester" could not be

"The Marine Steam Turbine."

[*To face page* 250.

Twenty-four Hour Endurance Trial at 22.5 knots.

Forward Engine-Room.	After Engine-Room.
230 lbs.	230 lbs.
98 ,,	93 ,,
0 ,,	2.25 ,,
19 in. vac.	...
180 lbs	...
53 ,,	50 lbs.
29.9 in. vac.	29.3 in. vac.
0	2 lbs.

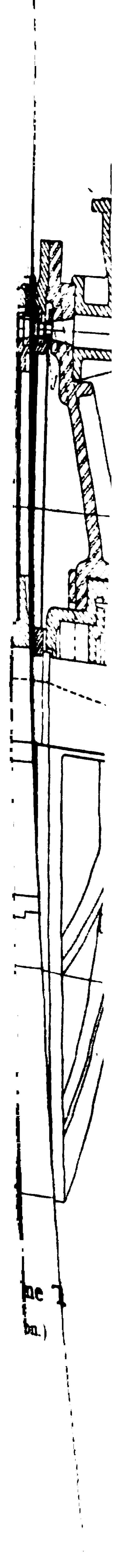
- - - -	185 degs. Fahr
- - - -	35 ,,
- - - -	64 ,,
ur - -	26 lbs.
- - -	.7 in.

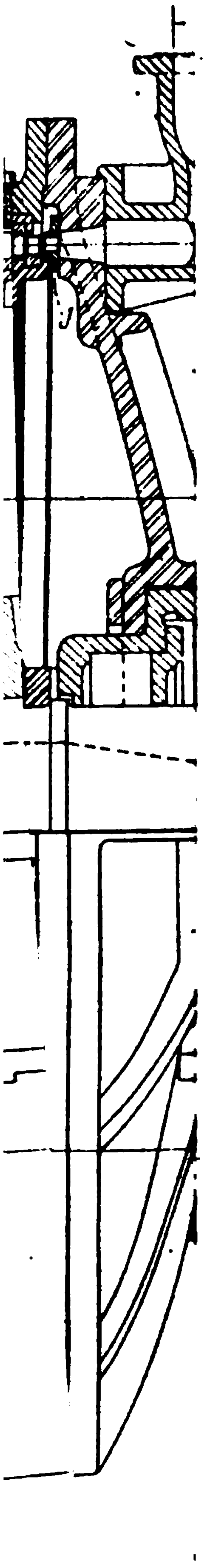
Full Speed Trial.

Forward Engine-Room.	After Engine-Room.
243.5 lbs.	246.7 lbs.
238.7 ,,	239.5 ,,
159.5 ,,	149.8 ,,
15.5 ,,	15 ,,
20 in. vac.	...
20 ,,	...
28 ,,	28 in. vac.
28.8 ,,	28.3 ,,

Port Propeller of U.S. Curtis Turbine Scout Cruiser "Salem."

Pitch, 8 ft. 8 in. Diameter, 9 ft. 6 in. Speed, 26·8 knots. Propulsive efficiency at 26 knots, 59.4 per cent. (referred to B.H.P.).





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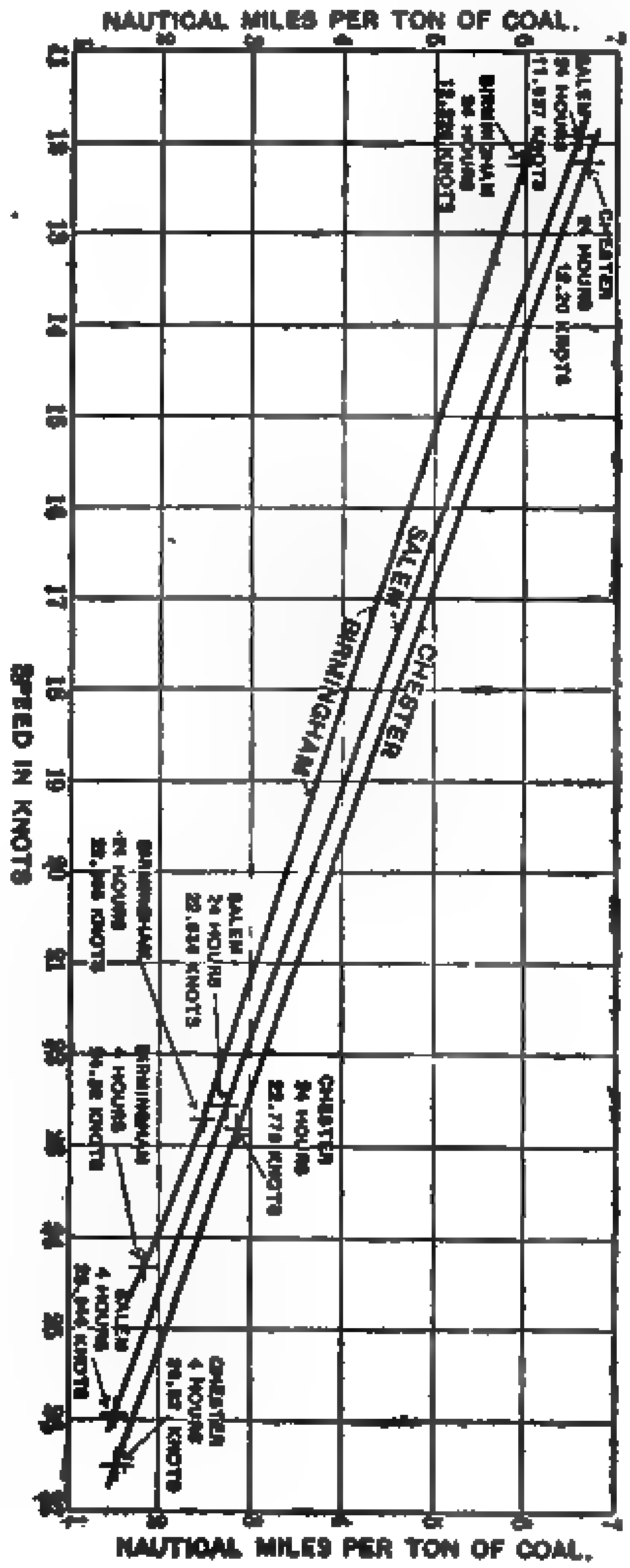
on.)

measured, owing to breakdown of two of the torsion mete
comparison of power and consumption at the progressive
therefore unknown. Otherwise the results are of particul
and it will be noticed that the "Chester" takes leading



CURVES FOR SCOUT CRUISERS, SHOWING NAUTICAL MILES PER TON OF COAL AT VARIOUS SPEEDS

as regards consumption and speed, with the "Salem" a go
At the same time it might be said that the reciprocati
vessel does not appear so economical as usually founc
practice, as 2.89 lbs. of coal per I.H.P. hour at 12 knots i

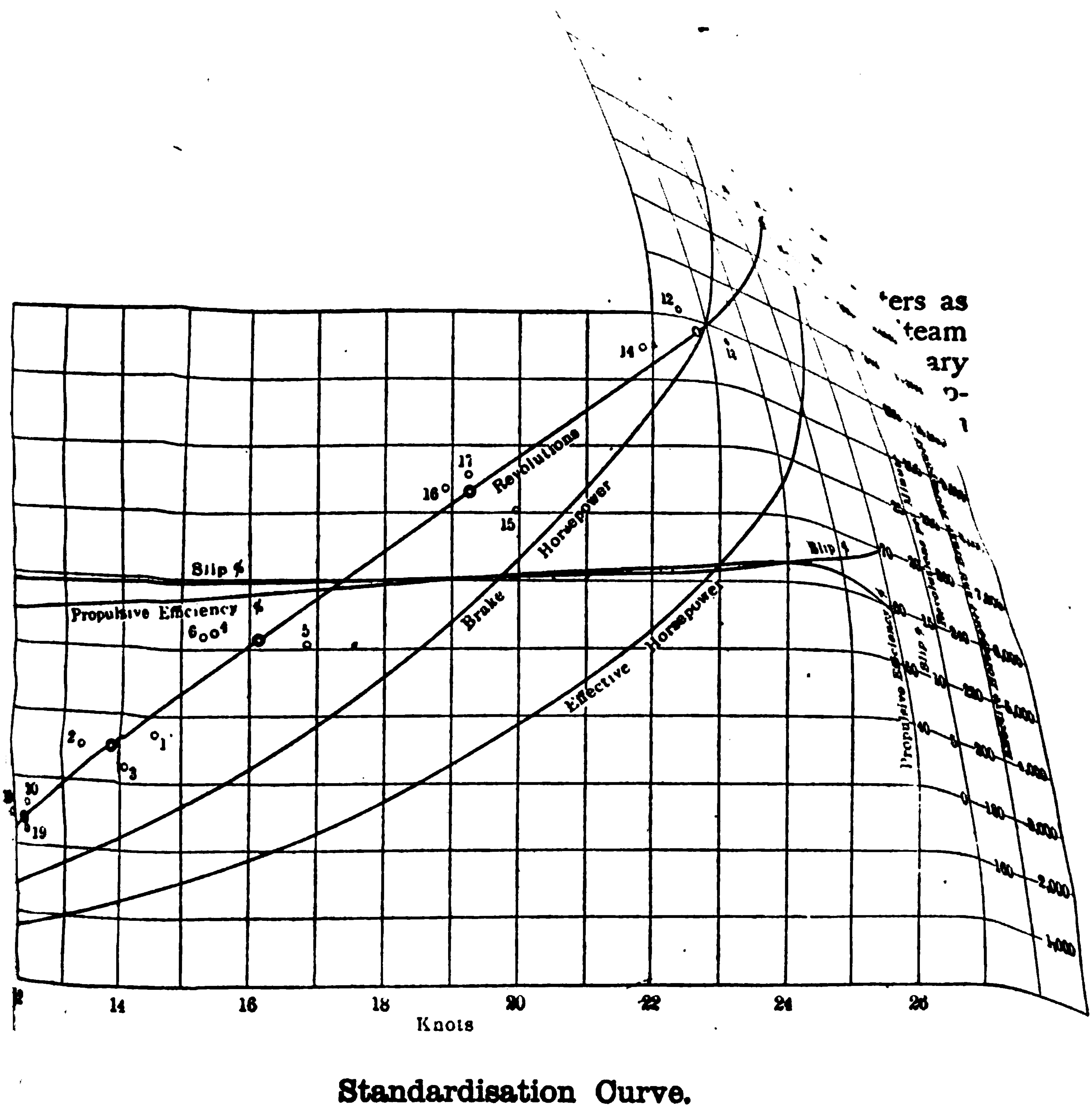
R

abnormal, even allowing that the engines were designed for th speed of 24 knots, as with reciprocating engines there very little difference in consumption per I.H.P. at either powers and speeds.

On referring to the trials of the U.S. cruiser scout "Sal be noticed that four sets of propellers were tried before thes the best results, were discovered. From this it may be assum propeller design is still in its infancy. It is, however, encouragin know that propellers of suitable and efficient design *can* be produc to meet the severe conditions imposed by high turbine revolution speeds, as, at maximum speed, the "Salem" turbines were running at nearly 400 revolutions per minute, and the propellers giving out a propulsive efficiency of 62.7 per cent. (referred to the shaft horse-power)—a figure abnormally high for turbine propellers, and perhaps never previously attained except in the case of reciprocating engines. Future improvement in propeller design will result in higher combined efficiency of turbines and propellers with a correspondingly reduced coal and steam consumption.

It is now common knowledge that a prominent Clyde firm is at present engaged in experimenting on turbines of the Curtis type, with the ultimate object of fitting them in a Government vessel at present under construction; the experiments will doubtless be directed towards the reduction in weight, which at present forms a rather serious objection to the Curtis type of turbine, and to improvements in consumption, as the economy is not what might be desired. The superiority of the Parsons turbine over the Curtis type lies chiefly in the reduced weight (turbine for turbine of a given power), and in reduced steam and coal consumption per horse-power hour, this latter being, of course, the crucial test from a commercial point of view. In the standard Parsons arrangement the total expansion of the steam is carried out in three turbines, one H.P. and two L.P., the H.P. terminal pressure (and the L.P. initial pressure) being usually from 12 to 20 lbs. gauge, or from 27 to 35 lbs. absolute pressure; whereas in the Curtis turbine the total expansion is effected in two single turbines of equal size, each receiving direct boiler steam independently and exhausting right down to the condenser pressure and temperature. Again, in the Parsons turbine, the H.P. casing is subjected to the full boiler pressure (less "drop"), while in the Curtis turbine the boiler steam expands and drops in pressure in the admission nozzles, so that the pressure on the casing is much less (see Table I., page 233). This expansion and pressure drop occurs at the initial end of each stage by means of the fixed nozzles through which the steam leaves one stage chamber to enter the next. It should also be noted that the steam in passing the guide blades of a Parsons turbine does work on itself to increase its own velocity, whereas in the Curtis type no such work requires to be done; the blade angles and openings being merely designed to guide or direct the flow of the steam from ring to ring of moving blades. This naturally results in a falling off in the velocity of the steam through the fixed and moving blades, and the required stimulus to the speed of the steam is obtained in the expanding nozzles of the succeeding stages, and this condition maintains throughout the turbine. An examination of the stage diagram (page 245) will make clear the differences in the blade angles and openings as compared with the Parsons system (page 15). Perhaps the chief reason for the excessive weight of the Curtis turbine lies in the necessity for having to continue the shaft or spindle right through the casing from end to end instead of finishing it off at the "wheels" as in the Parsons type. The blading system is, however, also to the heavy side, and the separate wheels to which the moving blades are fixed are necessarily somewhat massive in construction.

In the Curtis turbine the loss by blade tip leakage is very small as the leaking steam in each stage passes through the inlet nozzles of the next stage and is therefore utilised later on, the only actual loss being that of the last stage, where, however, the pressure is at a minimum. Leakage losses will occur, however, at the diaphragms or divisions between the successive stages where the revolving shaft passes through, although this has been reduced by screwing the holes in the diaphragms, which device, it will be noted, is similar in effect to the "fin" arrangement of the Parsons type gland rings.



Standardisation Curve.

U.S. Turbine Cruiser Scout "Salem."

Note.—Observe that at a speed of 23 knots the brake horse-power curve is cut by the vertical ordinate at 10,000, and the effective horse-power curve cut at about 6,250 horse-power, then, 6,250 ÷ 10,000 = 62.5 per cent. propulsive efficiency at this speed.

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SECTION IV.

TORSION METERS, &c.

The Bevis-Gibson Flash-light Torsion Meter.

THE following extracts from a paper, entitled "Torsion M
Applied to the Measurement of the Horse-Power of Marin
Turbines," by J. Hamilton Gibson, Esq., and read on 24th
1908 before the North-East Coast Institution of Engineers a
builders, describe the chief features of this flash-light type o
meter:—

"When a revolving shaft transmits power it always twists
throughout its length. In other words, the end at which the
applied moves slightly in advance of the end where the work
the amount of twist varying directly as its length, directl
moment of the load applied, inversely as the rigidity of the
and inversely as the fourth power of its diameter, the
reading—

$$\theta = \frac{10.2\ TL}{CD^4},$$

where θ is the angular displacement in radians, T = twisting
in inch-lbs., L = length of shaft in inches, C = the modulus of
and D = diameter of shaft in inches. The law holds good a
for all shafts which are not stressed beyond the elastic li
shafts are usually designed with a large factor of safety, i
that the amount of twist, or the 'torque,' as we prefer to c
very small. In propeller shafting, for instance, the torque
more than 1 deg. for 10 ft. of length, so that for a 12-in. shaft
cumferential displacement is only about $\frac{1}{8}$ in. at full power.

"Various methods and numerous instruments have been
to enable an observer to read off the torque of revolving shaf
such instruments are rightly termed 'torsion meters,' or, if self
ing, 'torsion indicators.'"

. . . .

Before applying any form of torsion meter to a shaft,

know its 'modulus of rigidity'—that is, how much it will twist with a given static load applied at the end of a lever of known length. This can only be done satisfactorily in the workshop, preferably on a long rigid lathe-bed. One end of the shaft is securely fixed and a twisting moment applied at the other end. To eliminate the effect of friction in the supporting bearing at the free end it is advisable to use two levers, one at either side, as shown in the illustration, and the loads are preferably applied by graduated spring balances. Two pointers independent of the load levers are secured to the shaft, as far apart as practicable, and the difference in the angular movement of these two pointers gives the true twist for that length of shaft. If the pointers are made 57.3 in. long from the shaft axis, their ends will describe 1 in. of arc for 1 deg. of twist, and a decimally-divided straightedge will then measure the twist to within $\frac{1}{100}$ deg., which is quite near enough for all practical purposes, and we can proceed to calculate the modulus of rigidity from the formula.

"Observe that a propeller shaft is subject to two distinct stresses. Not only is it twisted as between the engine and the propeller, it is also compressed longitudinally by the propeller thrust, the compressive stress being sometimes as much as 20 per cent. of the shear stress at the surface of the shaft produced by torsion alone. This compression augments the torque by an appreciable amount, which has been actually measured in numerous experiments, and may be taken roughly as 3 per cent. for hollow shafts and 1 per cent. for shafts which are solid. It might be considered sufficient to calibrate only one shaft in a multiple-screw vessel; but it is found that similar shafts, with identical tensile and elongation tests, have different moduli of rigidity, probably due to their varying elastic limits and some slight difference of homogeneity in the material. The only way, therefore, to ensure accuracy is to calibrate each shaft separately and to build up a power diagram, as shown in Fig. 1, for each.

"Another point to bear in mind is that a working propeller shaft is 'alive,' and this condition must be imitated as far as possible during calibration, by jarring the shaft with repeated blows of a mallet, so as to keep the mass in a state of molecular vibration. Otherwise the phenomenon of mechanical hysteresis, so marked in some static experiments, will obtrude itself and vitiate the results. Having established the true modulus of rigidity for each shaft, we may proceed to build up our power diagram based on the formula—

$$H = \frac{\theta D^4 N}{CL},$$

where H = shaft horse-power, θ = torque in degrees, D = diameter of shaft in inches, N = number of revolutions per minute, C = constant varying with the modulus of rigidity, and L = length of shafting in inches. In this formula we have all the elements for obtaining the shaft horse-power, and it only remains to ascertain the number of degrees of torque by means of a reliable and accurate torsion meter."

The Bevis-Gibson torsion meter is thus described:—

"Two blank discs are mounted on the shaft at a convenient distance apart. Each disc is pierced near its periphery by a small radial slot, and these two slots are in the same radial plane when no power is being transmitted and there is no twist on the shaft. Behind one disc is fixed a bright electric lamp masked, but having a slot cut in the mask directly opposite the slot in the disc. At every revolution of the shaft, therefore, a flash of light is projected along the shaft towards the other disc. Behind the other disc is fitted the torque-finder, an instrument fitted with an eye-piece and capable of slight circumferential adjustment. The end of the eye-piece next its disc is masked except for a slot similar and opposite to the slot in the disc. When the four slots are set in line, a flash of light is seen at the eye-piece every revolution, and if the shaft revolves quickly enough the light will appear to be continuous. This effect is apparent at anything over 100 revolutions per minute. At lower speeds the flash is seen to be intermittent, but this in nowise affects the accuracy and reliability of the result. At each end of the shaft, therefore, we have what is virtually an instantaneous shutter fixed, be it noted, directly to the shaft, and there is no connecting link or gear between the discs, either mechanical or electrical, except the beam of light which flashes once in each revolution clear through the two shutters. Let us suppose now the shaft to be transmitting power. One disc lags behind the other by a definite amount, and although three of the slots are still in line, the fourth slot, namely, that in the lagging disc, effectually blanks the flash and no light is seen at the eye-piece.

Flash-light Torsion Meter Results.

Engines, No. 1215.
Vessel, H.M.S. "Indispensable." Date, 31st November 1907.
Trial, official full power. At Clyde.

Shaft.		Run.	Steam at H.P. Receiver.	Revs. per Minute.	Reading.	Degrees Torque.	Shaft Horse-Power.
Starboard	Wing	IV.	180 lbs. sq. in.	710	5.90	1.26	8,300
	Centre			680	6.02	1.02	7,480
Port	Centre			687	4.64	1.21	7,520
	Wing			707	5.74	1.23	8,270
Mean revolutions - - - -				696	Total -		31,570

"This is where the function of the 'torque-finder' comes in. To

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ut ¼ in.
in., or
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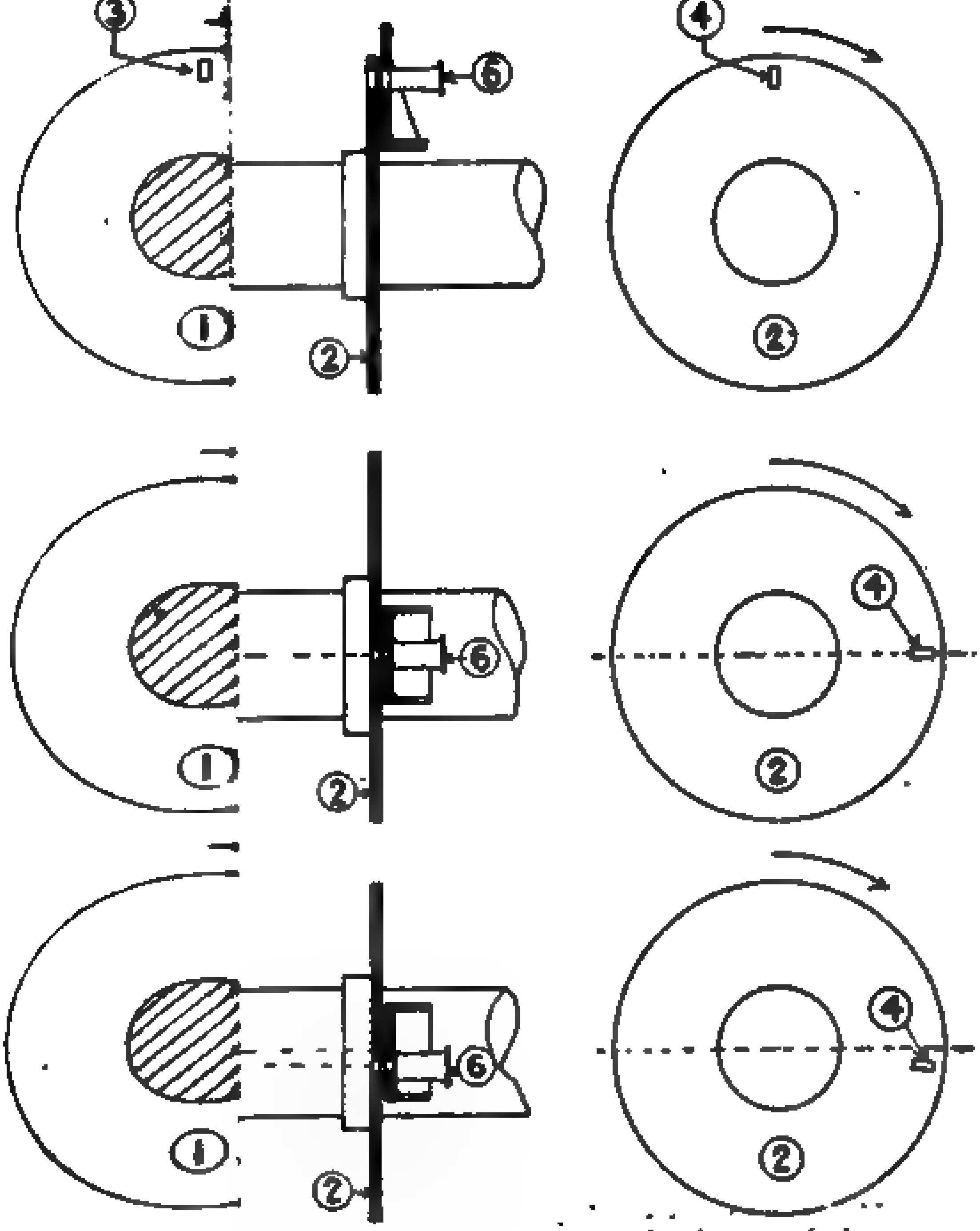


ıg along shaft.
out of position due to torque on shaft.

No so measure the torque or twist of the shaft.
the bottom plan.

[*To face page* 258.

der the light is visil
comes into view, att
ıd fades away as t

Turbine Shaft.	Degrees Torque.	Revs. per Minute.	Shaft Horse-Power.
Starboard L.P.	1.43	482.9	2,773
Centre H.P.	1.69	461.2	2,600
Port L.P.	1.37	472.8	2,600
Starboard L.P.	1.32	461.2	2,410
Centre H.P.	1.65	426.8	2,330
Port L.P.	1.24	457.3	2,200
Starboard L.P.	1.15	426.4	1,970
Centre H.P.	1.52	417.6	2,080
Port L.P.	1.13	418.9	1,910
Starboard L.P.	1.05	418.4	1,765
Centre H.P.	1.52	422.3	2,120
Port L.P.	1.02	415.5	1,670
Starboard L.P.	.21	198.6	162
Centre H.P.	.27	206.3	185
Port L.P.	.19	183.5	148
Starboard L.P.	.22	146.7	88
Centre H.P.	.21	171.4	87
Port L.P.	.13	144.8	82
Starboard L.P.	.07	46.3	13
Centre H.P.	.05	86.1	15
Port L.P.	.01	24.4	9.2

Fig. 2.—Actual Readings from a Flash-Light Torsion Meter and Horse-Powers taken during Trial Trips of a Turbine S Varying Conditions of Displacement and Propellers.

moves along the scale. If it were possible to gauge the where the light attains a maximum, that is the point th used. Failing this, however, use is made of one edge The finder is moved always in the same direction in taki

ed at the exact point where the light is cut off. So delicate ›e of sight that a movement of $\frac{1}{100}$ of a degree is sufficient ıe difference between light and darkness. A zero reading is :n the shaft is revolving idly, if possible at or near full speed, eading forms a base and is subtracted from any subsequent dings.

, suppose the shaft to be revolving idly. The finder is moved the light is just disappearing, and the vernier is seen to be ıt 0.53 degree. Now, suppose the shaft to be transmitting 'he discs have twisted relatively to one another, and no een until the torque-finder is moved the same amount.

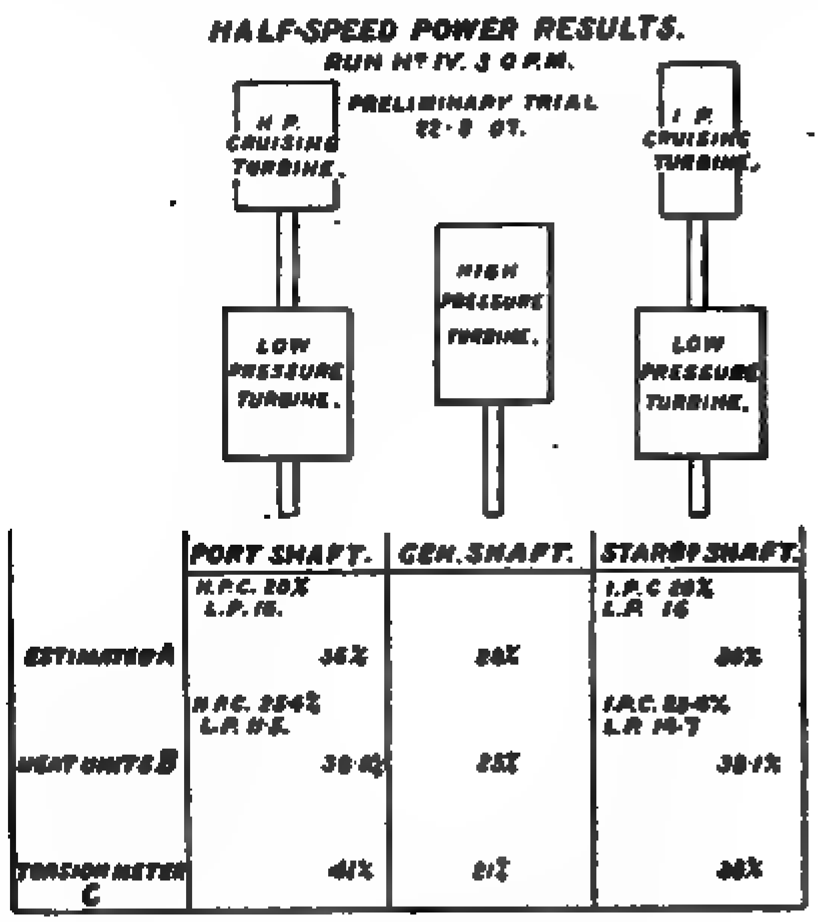


Fig. 3.—Percentage Distribution of Power for a Three-Shaft Turbine Installation.

ıcked up the light the finder is worked gently over until the ıst disappearing again. The reading is now 2.39 degrees. 'rom this the zero reading of 0.53 degree and we get the true mely, 1.86 degrees. Now, to apply our shaft horse-power ;. 1. We will suppose that the revolutions are 475 per The torque is 1.86 degrees, and finding the intersection of ›n the diagram the power is seen to be 3,620.

perhaps scarcely necessary to point out that the whole takes much less time than its description. Indeed, it is › produce the shaft horse-power on a trial trip immediately mination of each measured mile run, and to hand a slip to

the officer in charge similar to that shown in Fig. 2, containing all the information, in plenty of time for him to make any necessary adjustments before coming back on the straight for the next mile.

"Then again the distribution of power in a turbine installation can only be approximately estimated. The steam is turned into the high-pressure turbine and left to follow its own devious course through the successive turbines on its way to the condenser. At low powers it is sometimes found that the high-pressure turbine shows the most power, while for over-loads the lower-pressure turbines have the advantage. In Fig. 3 the percentage distribution of power is shown by three sets of figures—for a three-shaft turbine installation, including high pressure and intermediate cruising turbines. Set A shows the estimated or designed distribution, set B the calculated distribution from the pressure-gauge readings at the terminals of each turbine, and set C shows the actual distribution of power over the three shafts as ascertained by flash-light torsion meter readings.

"Referring again to Fig. 2, it will be seen that the starboard low-pressure turbine shows throughout the series more power than the port. Investigation showed that the blade-tip clearances of the two turbines differed slightly, and a further comparison proved that the percentage difference of clearance was just sufficient to account for the difference of shaft horse-power recorded."

Turbine Advantages over Cylinder Engines.

Some of the advantages of the turbine over the ordinary reciprocating engine may be stated as follows:—

1. Fewer working parts, as no piston or slide valves, piston rods, &c. &c., are required.

2. Steam applied direct from the boiler to the shaft without any intervening loss of power.

3. Less danger of breakdown, as it is almost impossible for the small vanes to become damaged, unless actual contact takes place between the blades and casing, or between the rings of rotor and casing blades. In addition to this, three separate lines of shafting are provided, so that if one became disabled, two are still left for running with.

4. Less weight of machinery for the same power.

5. Engines placed well down in the vessel.

6. Greater speed for the same consumption of coal at high speeds.

7. Machinery vibration practically eliminated.

It may, however, be stated that the turbine is most economical in running at high speeds, as the difference in consumption between, say, 20 knots and 10 knots is not proportional to the difference in speed.

This is due to the fact that the most economical speed of the turbine bears a certain ratio to the velocity of the steam, in addition to the fact that the loss by tip clearance or leakage is more proportionally at low speeds.

ith regard to the floor space occupied, there is not much to : between the reciprocating engine and the turbine. In space up in the vertical direction, the turbine has a distinct advantage. garding reliability, there is no reason why the turbine should be r to the reciprocating engine; and as to economy and speed rbine has shown itself in many ways superior, although at very eeds the reciprocating engine is superior in economy of steam; t this, however, the turbine requires less oil. The turbine, in its t state of development, is not suited for all classes of ships. the weight point of view the turbine is superior to the recipro- engine of the same power, particularly in the case of high-speed s, and possesses also the special advantage of being well balanced. may be some time yet before sufficient modification of the e allows of the adoption of this type of engine for the slow tramp steamer, but it is very probable that such modification me, in perhaps the near future.

Denny & Johnson's Patent Torsion Meter.

e following description of the appliance is reprinted from *ering*, 7th April 1905:—

: diagram, Fig. 1, shows the general arrangement of the apparatus lied to turbine-driven shafting. On the shaft, the torsion of which esired to measure, are fixed, at a suitable distance apart, two light tal wheels A and B. On each wheel is mounted, as shown, a ent magnet, the projecting pole of which is made V shaped, in order luce a dense and definite magnetic field at the point. Underneath gnets and set concentrically with the wheels and shaft are fixed two rs A and B, each of which consists of a quadrant-shaped piece of soft rried on a gun-metal stand provided with suitable levelling-screws. h piece of iron are mounted a number of separate but similar windings lated wire, there being a certain suitable number of windings per circumferential length of the iron. There is in conjunction with the rs a recording box, in which are mounted two series, A and B, of studs, around which are fixed scales. In connection with the series s two contact arms A and B are arranged, by means of which electrical tion may be made at will between any desired stud of a series and tact arm. There is in series A a stud for every separate winding nductor A, and in series B a stud for every separate winding in the r B, each stud being connected to its particular winding by means of a e wire, all the wires being contained in the multiple cables A and B. naining ends or returns of the winding on the inductors A and B are nected by means of two common wires (also contained in the cables) to the contact arms A and B respectively.

luded in each of these two circuits is a variable resistance (by means h the strength of the current flowing in the circuit may be adjusted red) and one winding of a differentially-wound telephone receiver. le A is divided into six equal parts, there being six separate windings inductor A, and thus six studs in the series A; the length of five sions of the scale thus represents the circumferential length occupied

by all the windings on the inductor, each subdivision representing the distance between the neighbouring windings, which is usually 0.2 in. The scale B is divided into fourteen equal parts, there being fourteen separate windings on the inductor B, and thus fourteen studs in the series B. The length of thirteen subdivisions of the scale, as before, represents the circumferential length occupied by all the windings in the inductor, the distance between neighbouring windings being represented by one subdivision of the scale; the usual distance between neighbouring windings in inductor B is 0.02 in. The wheel in connection with the inductor at the turbine end of the shaft is set so that the magnet fixed to it is exactly above one or other of the two end windings in the inductor.

The correct end winding to set the magnet to is that one from which the

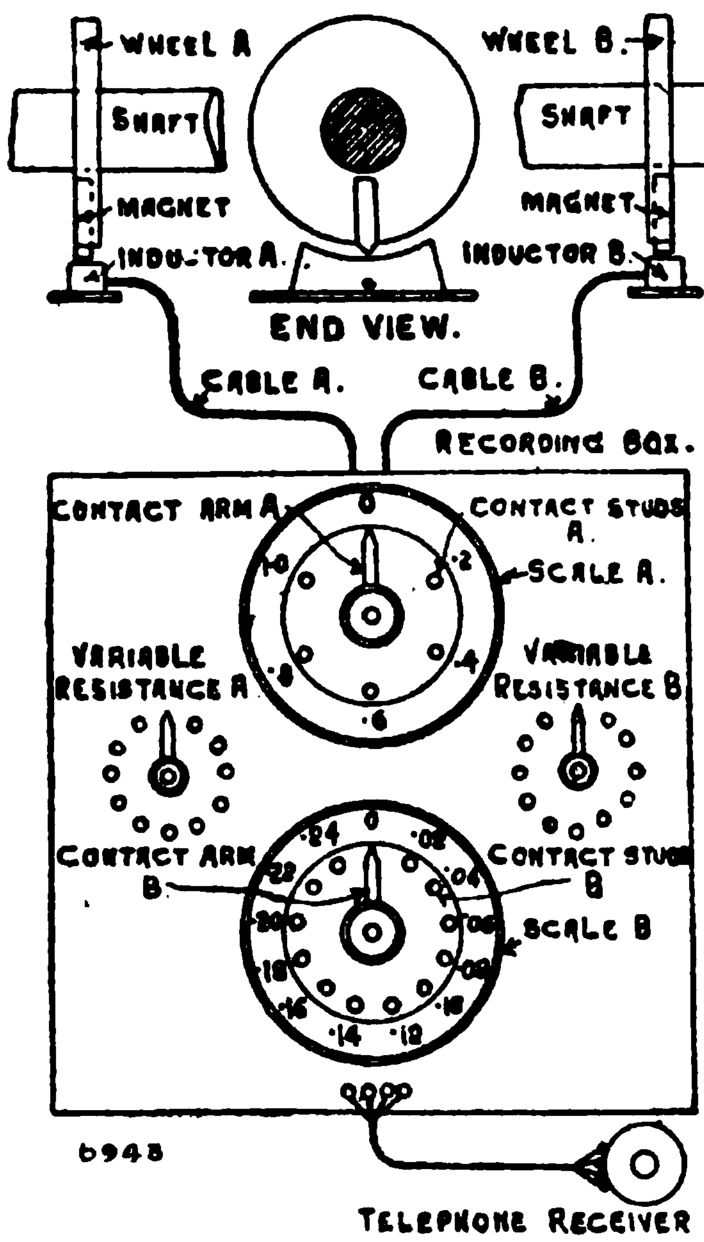


Fig. 1.

magnet will travel towards the other end winding when the shaft rotates. The wheel in connection with the other inductor is also set so that its magnet is exactly above one of the two end windings in the inductor, the correct end winding to set the magnet to in this case being that one from which the magnet will travel in the opposite direction to the other end winding when the shaft rotates. To facilitate the accurate and easy setting of the magnets above their respective windings, lines are cut in the tops of the inductors exactly above the end windings, and the magnets are set to these lines. When the shaft rotates without transmitting power, a current of electricity is induced in the end or zero winding of each inductor, the contact arms being first placed in contact with the end or zero stud in each series. These two separate currents both traverse their respective circuits, passing in each case from the inductor winding in which they are induced to the

respective zero studs to which these windings are connected, thence by way of the respective contact arms, resistances, and telephone receiver windings back to the inductors again. The connections to the receiver are so arranged that the effects of the two separate currents flowing re in opposition, and thus neutralise each other's effect on the hen the strengths of the two currents flowing are exactly equal at instant. By means of the variable resistances in each of the circuits nts are made equal in strength, and then so long as the shaft no power, and is thus subject to no torsion, no sound will be heard ng at the receiver, since the currents induced in the zero windings ductors have been equalised, and are both induced at exactly the ant. When transmitting power, the shaft is subject to a certain r twist, which causes the zero winding of the inductor next the r engine to be excited in advance of the other by the amount of n of the shaft; a loud ticking sound will then be heard in the s the currents no longer neutralise each other.

ct arm B is then shifted from stud to stud, until the position of ilence in the receiver is once more obtained. When this position the reading on the scale B, opposite the contact arm, represents mferential measurement of the angle of torsion of the shaft at the the inductor windings. A current equal in strength at the same that induced in the zero winding of inductor A is now being induced inding of inductor B which is in connection with the contact e scale reading thus represents the displacement of one magnet with the other due to the torsion of the shaft. In the event of the ing found to be too great to be measured on scale B alone, contact hifted from stud to stud until a reading can be obtained on scale rsion reading being equal to the sum of the readings on scales The reading corresponding to any large displacement of one elatively to the other is thus easily obtained by the combined scales A and B.

Type A Instruments.

ype has been designed for measuring the torsion of shafts having urning moments, *e.g.*, shafting driven by turbines, electric motors,

ix the Apparatus.—A set of instruments as applied to a shaft the following items:—(1) Two light gun-metal wheels, each fitted ermanent magnet; (2) two inductors; and (3) a recording box. ing up the apparatus it is first necessary to determine what length ilable shafting will be required to give a reasonably large torsional The power to be transmitted by, and the revolutions of, the shaft ximum load are first approximately calculated. The diameter of the en measured, and the inductor constant noted. (This constant is the ius in inches at which the torsion of the shaft is measured; and it is n all inductors.) Having made these observations, the length of feet required to give approximately 1 in. of reading is obtained by e formula,

$$\frac{1.53 \times R \times d^4}{C \times H.P.} = \text{Length.}$$

Where R = revolutions per minute of the shaft, C = inductor cons diameter of shaft in inches, H.P. = horse-power (approximate), and constant which takes account of the figure 140, a constant which ascertained from a great number of static calibrations of shafts, and an expression of the torsional resistance of solid iron and steel shaf the case of hollow shafting the torsional strength can best be f calibrating the shaft (see page 267), but it is probable that the strength is the same as that of a solid shaft of the same diameter m resistance due to a solid shaft which would fill the bore, and, in the of calibration, the error due to this assumption is probably a r quantity.

Having found the length of shafting which will give 1 in. of rea length is selected if available, and tne wheels fixed to the shaft desirable to select the greatest length of shafting available, prov length chosen is such that the full load reading shall not exceed being advisable to have the remainder of the scale in reserve in the the power, &c., having been underestimated. In most cases it advisable to sacrifice a short length of the available shafting in o the wheels may be fixed close to bearings (as shown in the illustrati allowing the wood or iron soles on which the inductors are to be mo be bolted to the bearing stools. The advantage of this is that any that may be present when the shaft is running affects the shaft and stools equally. If necessary, however, the inductors may be fixe other position, provided that a good solid base is provided. After t wheels in place, the inductors should be screwed to the soles, concentrically with the wheels. The setting is done by means of tw pieces supplied with the instrument. These are inserted between of the inductor and its wheel, the levelling screws in the base of the being adjusted until the setting pieces both fit tightly and bear th their entire length. The locking nuts in the base are then tight and the setting pieces removed. The top or covering plate of each has two lines or grooves cut across it, and the wheel in connection inductor fixed at the turbine or engine end of the shaft must be adjuste shaft until the magnet on that wheel is directly above the line fro it will travel, and towards the other line when the shaft rotate *necessary to set the magnet above the correct line (which is the corresponding to the direction of rotation of the shaft) or no rea be taken.* The magnet is arranged in a tight fitting groove in the that by removing the bridge pieces which hold it in position the ma be lowered until its sharp edge rests exactly in the zero line. Th should then be clamped tight to the shaft, and the magnet raised a in position with its edge about $\frac{1}{16}$ in. clear of the top of the indu gauge-marked air gap is supplied for the purpose of testing the cleara

In addition to the two zero lines there is also a line marked circ tially on the top of the inductor, and when setting the inductor this lin be made to coincide with the lines which are cut on the magnet. T inductor is set and fixed in a similar manner; but in this case the must be set to the zero line at the opposite end of this inducto inductors must be so placed that the connections are on the same both on the right or both on the left hand side of the inductors, in o the induced currents of electricity may be in the proper directions,

trouble of altering the telephone connections thus avoided. The cables should now be fixed to the inductors. The plugs attached to the ends of the cables are inserted into the sockets on the inductors, care being taken to see that the numbers on the plug and socket carriers correspond. The plugs must all be so inserted that the arrows on the plug carriers always point from the lines to which the magnets are set and in the direction of the other line. (This is essential.) The cables are not the same length. The longer one is intended to go to the inductor farthest away from the recording box (care should therefore be exerted when fixing the inductors to see that the inductor intended for use with the long cable is placed at the proper end of the shaft). This will probably be the after-end in all ships. The recording box should be placed in a reasonably quiet place, preferably a room as far removed from the noise of the machinery as is possible. The cable connections are then made to the recording box, and as the plugs and sockets are numbered no mistake should be possible.

To Take a Reading.—First see that the *resistance* contact arms and the *main* contact arms in the recording box are both at zero. The telephone receiver is then placed to the ear of the observer, and when the shaft is revolving a ticking sound will be heard. Shift the *main* contact arm of the $\frac{1}{100}$ths scale from stud to stud until a position is found where no tick is heard, or where the sound is reduced to a minimum. The torsion reading will then be found on the scale opposite the *main* contact arm. Should the tick not cease entirely but appear to be equally low on each of two neighbouring studs, the correct position of the arm is between those studs. When no reduction of the ticking noise can be obtained by moving this arm, it is evident that the torsion of the shaft is greater than the range of the $\frac{1}{100}$ths scale. The arm of the $\frac{1}{8}$ths scale must then be shifted from stud to stud until a reading can be obtained on the $\frac{1}{100}$ths scale. The torsion reading will then be the sum of the two scale readings. We have assumed that no balancing or equalising of the currents by resistance was required, but in practice this will usually require to be done before readings can be accurately taken. The equalising is accomplished as follows:—First take a trial contact arms reading as described above, and then move one of the resistances (the correct one being found by trial) until a position is found where the sound in the receiver almost or entirely disappears. It may be necessary to slightly alter the trial reading, as well as the resistance, before silence is obtained, and in cases where the equalising cannot be satisfactorily accomplished this will no doubt be due to the receiver being too sensitive for the rate of revolution. This can be remedied by adjusting the tension screw at the back of the telephone receiver until the desired degree of sensitiveness is obtained. Having once found the correct position for the resistance contact arm, no further adjustment will be necessary for a particular shaft or set of shafts. When applying the instrument to another shaft, or in cases where the revolutions of the shaft under trial vary greatly, it may be desirable to slightly readjust the resistance contact arm and also the receiver. With turbine-driven shafting in ships, where the rate of revolution is sufficiently high, a very good check may be applied to the accuracy of the zero setting of the inductors in the following manner:—After the day's trial is over, let the steam be shut off one turbine, and while the ship is driving ahead with the remaining turbines note the reading of the idle shaft, which will continue to revolve due to the action of the water on its propeller. The friction of the shaft bearings and of the turbine

may be assumed to require a relatively negligible amount of power to be transmitted by the propeller along the shaft, so that the reading thus taken should agree with the zero reading. To find the zero reading by this check method, the main contact arms of the two scales are placed opposite the letters z, when no sound should then be heard in the receiver if the inductors have been properly set. If a sound be heard, the arm of the $\frac{1}{100}$ths scale must be shifted until the tick disappears, or until the sound is reduced to a minimum. If the reading thus obtained is less than the reading at z, then

Recording Box.

the difference between the two readings is the amount of zero error, and must be added to the whole of the readings previously taken. If, on the other hand, the reading is greater than that at z, then the difference between the two readings must be subtracted from the whole of the readings previously taken. The foregoing test need only be applied, however, in the event of doubtful readings being obtained, as the set zero will always be correct if the inductors are carefully set up.

To Calibrate the Shaft.—Previous to being placed in its position the

shaft should be calibrated. This is done by coupling the various lengths together to include the length over which the torsion is to be measured, and ng it on bearers and twisting it by means of weights in the following —Clamp one end of the shafting rigidly in position so that it cannot fix a stout lever to the other end. On to the same end of the shaft ter 6 or 7 ft. long, so that it will move over a paper scale. Weights en be attached to the outer end of the lever, and the deflection of er in inches noted as the weight is increased until a set of readings up to the maximum calculated torque are obtained. A curve is wn, the ordinates being foot-pounds and the abscissal torsion in

Find the Shaft H.P. from the Torsion Reading.—From ation curve the foot-pounds corresponding to any torsional reading aft taken by the torsion meter may be easily found. The torsion dings must first of course be made to correspond with the length of l shaft and to the calibration radius by means of the formula,

$$\frac{r \times l \times L^1}{C \times L} = K,$$

= torsion meter reading in inches, l = length of pointer in inches, th in feet of calibrated shaft, C = inductor constant, and L = length shaft from which the torsion meter reading is obtained, K = reading meter corresponding to the curve.

ng found from the curve the foot-pounds corresponding to any reading torsion meter, the horse-power transmitted is then found by means mula,

$$\frac{F \times R}{5255} = \text{horse-power transmitted by the shaft,}$$

= foot-pounds, R = revolutions per minute of the shaft, and 5255 is a

e case of solid shafts of iron or steel, instead of calibrating the shaft ant 140 (which figure represents the average torsional resistance of steel shafts as found from numerous calibrations of shafts) may be as correct, and the horse-power transmitted by the shaft found by the formula,

$$\frac{1.53 \times r \times d^4 \times R}{C \times L} = \text{horse-power.}$$

bove formula, d = diameter in inches of the shaft, R = revolutions ite of the shaft, C = inductor constant, L = length in feet of shaft, ng of torsion meter in inches, and 1.53 is a constant.

w shafting should be calibrated, as the torsional resistance varies bore, which may not be sufficiently uniform to permit of very calculation.

strongly recommended, however, that all shafting, whether hollow or ould be calibrated if at all possible, as very great accuracy is then When the constant 140 is used in place of calibration the average error is about 1 per cent.

it Improvements in Turbine Construction and Design.

ling.—The blading now generally adopted is that of the

"segment" type, which allows of the simultaneous construction of the rotors and casings with the blade fitting, as the various segments of blades and "packers" only require to be placed in position and caulked down into the grooves to complete the work. This effects a considerable saving of time in the work of construction. The Admiralty practice of thinning away the blade tips to prevent damage should wear-down occur is usually carried out in all long blades at least, and often throughout the blading.

Wheels.—In large turbines the wheels are now usually made of steel forgings in place of steel castings as formerly. When large wheels are of cast steel special allowance is made for shrinkage by having a cut made in the rim, which is afterwards filled in by a fitting strip when the metal has cooled down and set. This obviates the setting up of severe strains due to the casting cooling and contracting unequally owing to varying thickness, &c. Brass wheels are also being experimented with.

Glands.—Various methods of gland fittings have been experimented with during the past two years, and the type now adopted, and evidently standardised, consists of four Ramsbottom rings in combination with twenty or more radial fins, the gland case itself being in two halves. In the H.P. turbines steam is admitted to a pocket between the fins and rings during heating up, but when running the same pocket acts as a "leak-off" to the 3rd expansion of the L.P. turbines.

In the L.P. turbines the pockets referred to admit steam at a low pressure, and do not act as "leak-offs," as these are not required with L.P. turbines.

A type of gland made up and fitted in three segments has been fitted in a few cases (notably in the turbines of the T.SS. "Cairo" and "Heliopolis"), but owing to the labour and time required in overhauling and putting together again, this type has wisely been discarded, the arrangement being much too complicated for practical operation. It may be stated that the problem of the ideal gland has yet to be solved, as the present arrangements do not give complete satisfaction.

Dummies.—The dummy rings are now much more undercut than formerly, resulting in a reduced thickness of metal presented for contact should longitudinal wear or displacement of the rotor take place. This tends to less risk of serious damage resulting from contact of the grooves and rings.

In a few cases which came under the observation of the writer the dummies had been in actual contact when running, but no serious damage ensued, the only evidence of metallic contact being deep circumferential marking of the grooves in the dummy piston. It is also worthy of note that the dummy "leak-offs" are now abandoned, as it is considered that they merely form a passage through which

steam will find its way unnecessarily from one point of the turbine to another. As trouble has been experienced by corrosion occurring in the dummies, due to galvanic action between the brass rings and cast-iron case, it has been suggested to make the dummy casings and pistons all of brass instead of cast iron, but the cost of this material forms an important objection, and, in any case, it is likely the corrosion would still take place at some other position.

Oil and Water Service.—The oil service is now much improved in many ways, the oil circulating through sight glasses fixed to the bearings, which afford a reliable index to the oil flow. In some cases small cocks are fitted at different positions on the bearings, and these have a small hole drilled through the shell so that when the plug is turned round the oil spurts out if present. In the most recent system the oil is cooled independently before entering the bearings, the water service to the bearings having been done away with. The oil is also filtered in special gravity tanks before or after cooling, and to allow for temporary stoppage of the oil flow from the oil pumps, the " reserve tank " system has been adopted in some cases, but cannot be said to be altogether satisfactory, as the reserve supply of oil in the overhead tanks would only suffice for a very few minutes' actual running of the turbines should the pump supply be cut off.

Running and Upkeep of Turbines.

An important point in favour of turbines is the small amount of repairs required, as owing to the small number of bearings and moving parts, and there being practically no wear of bearings, repairs are reduced to a minimum. To attain this result and to give best economy in running turbines, the following advice may be of use. In dealing with the bearings, care should be taken that all the oil supplied to bearings is carefully strained and filtered. In most of ships filters are fitted on the supply pipe to bearings. The filtering cloths in these filters should be kept as fine as possible, and to ensure quickness in overhauling, a spare set of filtering grids should be kept. The filter is opened, and dirty grids taken out, and clean ones put in. The dirty ones can now be thoroughly cleaned, ready for reinstatement. The oil should be kept at as steady a temperature as possible, and oil in suction tank kept up to requisite level.

A wear-down gauge is usually supplied similar in purpose as in a reciprocating engine. This should be used frequently to ascertain if any wear has taken place in bearings.

The adjusting or thrust block should be examined to see that there has been no wear on the rings, and if so, then it is necessary to readjust same in relation to the rotor dummy. It will be seen that the finer the dummy adjustment the more economical the turbines will be. It is not advisable to bring dummy too close, as disaster may occur, but a dummy can be run at .01 clear of blades on casing dummy, and this is about as close as it is safe to go. When the

clearance is fined down to this extent, it is advisable to do this setting after turbine is warmed up. Previous to heating up, the rotor should be pressed aft until the dummy is .030 clear of blades. After turbine is thoroughly warmed up, the rotor can be brought forward again and clearance fined down to .01 or .015 as is thought safe. This is ascertained from the finger piece. The clearance being fine, less steam will escape to the interior of the rotor, instead of doing work going through the blades. Turbines are designed so that the propeller thrust will be balanced by the steam thrust on the blades, and if this balance is perfect, then the shaft collars will not touch the rings on the adjusting bush, but will be floating clear. After machinery is stopped and cooling down, it is advisable, if working with a fine dummy clearance, to press the rotor aft, until dummy is from .030 to .040 clear, as in the event of shaft cooling down before rotor, it will draw rotor forward and considerably decrease the dummy clearance. This operation can be avoided by keeping a gentle flow of steam acting on the shaft, at the forward gland, so as to allow the rotor and casing to cool before shaft is cooled. In running turbines, care must be taken to avoid priming of boilers. This is a very important matter, as the passing of boiler water into the interior of the turbines may not do any material damage to the blading at the time, but if the boiler water is salt, then salt deposit will take place, and this, becoming solidified, will throw the rotor out of balance. Salt and other sediment being carried into the turbines will get in between the blades, and may result in closing up the blading in the 1st and 2nd expansions, with the result that the steam will not be doing any effective work, but will flow past the tip of the blades, and cause a higher pressure to show in the receiver. When starting up, the drains on the H.P. casing should be open, and the air pump drawing from bottom of L.P. casing so as to clear any water that may gather, before the turbines are thoroughly heated up.

Attention should be given to revolutions of each turbine, and these should be made to correspond as nearly as possible. It may be found that one L.P. is running faster than the other; in this event, the valve on the L.P. main steam inlet of the turbine which is running fastest should be closed a little so as to cause an increased amount of steam to pass to the turbine which is running slowest. The turbine casings are lifted for survey by Board of Trade once every twelve months in passenger ships, and a thorough examination should then be made of blading to see that no fraying of blades is taking place, and of interior of rotor drums to see that no corrosion or salting up is taking place.

Combined Reciprocating Engines and Turbines.—The most recent practice in mercantile steamers is the combination of reciprocating engines and turbines for vessels of moderate or low speeds. Several steamers of this class are at present under construction, and the arrangement consists of two wing triple or quadruple expansion

engines exhausting into a central low pressure ahead turbine, driving a third shaft and propeller, the revolution speed of the centre turbine shaft being much higher than that of the wing shafts. An alternative design is that of two wing low pressure turbines and one centre reciprocating engine. The turbine for the Dominion Line steamer now under construction by Messrs Harland & Wolff is 10 ft. 4 in. in diameter over blades, and is supplied with 6 expansions, the blade heights varying from 3 in. to 10 in.; while the turbine for the N Z. Shipping Company steamer has a diameter of about 7 ft. 6 in. and is fitted with 5 expansions, 3 of increasing blade height and 2 of uniform height.

In the steamers mentioned the engines are arranged to be run as follows:—

(1) Boiler steam to both H.P. cylinders, and exhaust from these to turbine, the exhaust from the turbine being divided and led into two separate condensers.

(2) Boiler steam to both H.P. cylinders, and exhaust from these to condensers direct, the turbine being then cut off. This is required when running astern as the turbine is for ahead running only, and may be used for ahead running with two propellers only.

(3) Boiler steam to either H.P. cylinder, and exhaust from L.P to centre turbine, then into one condenser only.

These combinations are obtained by the use of "change valves" fitted on the L.P. exhaust pipes, and by large butterfly valves fitted in the turbine exhaust branches. The change valves admit the reciprocating exhaust steam of either side to the turbine, or to the condenser as required, and the large valves in the turbine exhaust pipe shut off the condenser on either side as may become necessary should one reciprocating engine require to be disconnected through breakdown.

Benefits of the System.—As, broadly speaking, the economy of the reciprocating engine depends chiefly on high-pressure steam, and the turbine on low-pressure steam, the judicious combination of the two ought to result in higher efficiency results. The theoretical heat gain may be figured out as follows:—

L.P. terminal pressure = 8 lbs. absolute.
Condenser back pressure = 2 „
Assume dryness fraction at 8 lbs. absolute = .82.
„ „ „ 2 „ = .78.

Then on referring to the steam table, page 3, the latent heat at 8 lbs = 985.7 B.T.U., and the latent heat at 2 lbs. pressure = 1025.8 B.T.U. The temperature at 8 lbs. = 182.9° Fahr., and at 2 lbs. 126.1° Fahr., and adding 461 for absolute temperature to each of these we obtain $182.9 + 461 = 643.9$ and $126.3 + 461 = 587.3$.

Then referring to page 34 we find the formula for work done by adiabatic expansion.

Therefore, $985.7 \times .82 - 1025.8 \times .78 + 643.9 - 587.3$
$= 808.274 - 800.124 + 643.9 - 587.3$
$= 808.274 + 643.9 - 800.124 - 587.3$
$= 1452.174 - 1387.42 = 64.75$ B.T.U. per lb. steam.

Assuming a steam flow of, say, 1882 lbs. per minute, and an efficiency of 55 per cent. for low-pressure turbines—

Then, $\left.\text{Equivalent I.H.P.} = \dfrac{64.75 \times 1882 \times 778 \times .55}{33000}\right\} = 1580$ (nearly) horse-power.

The turbine is therefore most effective in dealing with steam of a pressure which cannot be utilised with benefit in a triple or quadruple expansion engine, owing more particularly to the huge volumes involved, and requiring increase of weight, space, and frictional losses.

It is understood that the L.P. exhaust pressure to the centre turbine will not be more than 7 or 8 lbs. absolute. This will produce a difference in the usual L.P. cylinder diagram cards, bringing up the exhaust line to a position much nearer the atmospheric line than usual.

The loss of work energy so represented by the reduced indicator card area in the L.P. engine will be more than balanced by the increase of power developed in the turbine.

The economical result of the combination arrangement is, to all appearance, beyond question, and may in time, with suitable improvements which experience suggests, prove adaptable for the usual tramp steamer speed of from 8 to 10 or 11 knots.

An innovation has been made in the case of the turbine glands, which, instead of the frictionless steam packing hitherto adopted, have in one case been changed for the usual marine type of piston rod gland, consisting of rack and pinion screwing-up gear with soft packing inside, this type of gland being fitted at both ends of the turbine

Theoretical Data for Combination System of Reciprocating Engines and Turbines.

H.P. initial pressure	185 lbs. (gauge).
Turbine initial pressure	12 in. vacuum, or 9 lbs. absolute.
Condenser	28 in. vacuum, or 1 lb. absolute.
Steam flow per hour	52,000 lbs.
Reciprocating engine over-all efficiency (assumed)	.75.
Turbine over-all efficiency (assumed)	.54.

Then,

$185 + 15 = 200$ absolute initial H.P. engine pressure.

200 lbs. $= 381.7°$ temp. and $381.7° + 461° = 842.7°$ absolute temp.

200 lbs. $= 845$ B.T.U. of latent heat.

9 lbs. absolute $= 188.3°$ temp. and $188.3 + 461 = 649.3$.

9 lbs. absolute $= 981.9$ B.T.U. of latent heat.

1 lb. absolute $= 102.1°$ temp. and $102.1 + 461 = 563.1$.

1 lb. absolute $= 1042.9$ B.T.U. of latent heat.

Dryness fraction : 1 for initial pressure of engine, .845 for initial pressure of turbine, and .771 for terminal pressure of turbine.

Then, heat drop per lb. steam in reciprocating engine working between 200 lbs. initial and 9 lbs. terminal pressure, and assuming adiabatic expansion throughout

$$= 845 \times 1 - 981.9 \times .845 + 842.7 - 649.3$$
$$= 845 - 829.70 + 842.7 - 649.3 = 208.7 \text{ B.T.U.}$$

Heat drop per lb. steam in turbine working between 9 lbs. initial pressure and 1 lb. terminal pressure, also adiabatic expansion,

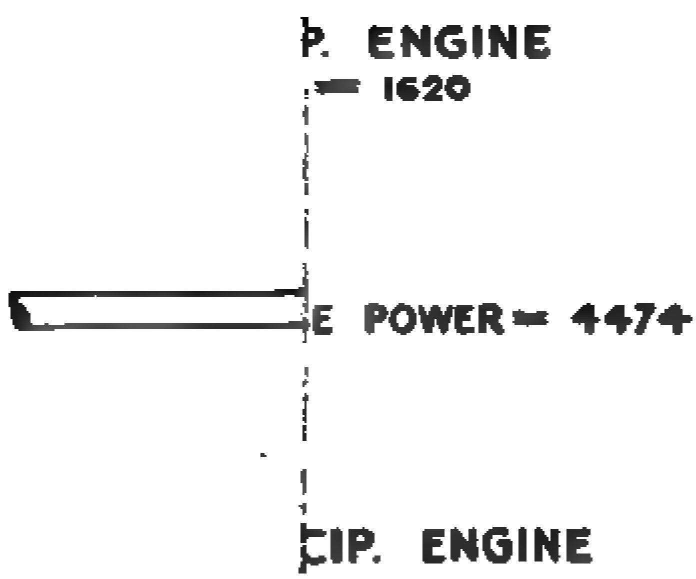
$$981.9 \times .845 - 1042.9 \times .771 + 649.3 - 563.1 = 111.9 \text{ B.T.U.}$$

Steam flow per hour through each reciprocating engine	$= 52{,}000 \div 2 = 26{,}000$ lbs.
Horse-power developed in each engine	$= \dfrac{26000 \times 208.7 \times 778 \times .75}{60 \times 33000} = 1599.$
Total horse-power developed in both engines	$= 1599 \times 2 = 3198.$
Steam flow per hour through turbine	$= 52{,}000$ lbs.
Horse-power developed in turbine	$= \dfrac{52000 \times 111.9 \times 778 \times .54}{60 \times 33000} = 1234.$
Combined horse-power -	$= 3198 + 1234 = 4432.$
Lbs. steam per horse-power hour	$= 52000 \div 4432 = 11.7$ lbs.
Lbs. coal per horse-power hour (evaporation 9 lbs.)	$= 11.7 \div 9 = 1.3$ lbs.

Appended will be found an extract of a paper by the Hon. C. A. Parsons and R. J. Walker read before the Institution of Naval Architects, 9th April 1908, entitled "The Combination System of Reciprocating Engines and Steam Turbines," and in which various data are given relating to the power and consumption expected by this combined arrangement:—

"It may be said that perhaps the most important field for the combined system of machinery as applied to marine propulsion is for those installations where the designed full speed of the vessel falls below the range suitable for an all-turbine arrangement, the reciprocating engine working in the region of pressure drop where the conditions are best suited for it, and the turbine utilising that portion of the expansion diagram which the reciprocating engine is not able to utilise efficiently. It is generally well known that an all-turbine arrangement has not been advocated by us for ships where the designed speed falls below 15 or 16 knots, excepting in some special cases, such as yachts; and for vessels of moderate or slow speed the combination system of machinery appears to be eminently suitable.

"In a good quadruple reciprocating engine, the steam is expanded down to the pressure of release, about 10 lbs. absolute, and gains in economy as the vacuum is increased up to about 25 in. or 26 in.;



(1) Chat to turbine or to condenser.

[To face page 274.

whereas in a turbine it is possible to deal economically with very low-pressure steam, and to expand this low-pressure steam to a low ·ıte pressure corresponding to the highest vacuum obtainable in ıe practice.

Fig. 4 shows in diagrammatic form the advantage of the combined n of machinery. The total area of the diagram represents the mum energy that could be obtained, theoretically, from the steam vere expanded down to the pressure in the condenser. The area sed by lines A, B, C, D, and E shows the theoretical maximum y realisable in a quadruple engine from 200 lbs. pressure to 26 in. ım, and the area cross-hatched the additional energy that can be ɛd in a turbine, but which cannot be economically used in a ·ocating engine.

In a combination system, the most suitable initial pressure for the ıe, or the dividing line between the reciprocating engine and the ıe, will greatly depend upon the conditions of service of the par-r vessel taken. The reciprocating engine, or engines, could be ned to exhaust at a pressure of between 8 lbs. and 16 lbs. absolute, ɛn at a slightly higher pressure, if necessary, to meet the con-ıs required. From an estimate of the theoretical efficiency under ırious conditions of pressure, as set forth in the following table, ıld appear, apart from any practical considerations, that there is ng to choose between an initial pressure at the turbine of between and 15 lbs. absolute, any pressure within this limit appearing to he most economical result.

	Initial Pressure Turbine.	Reciprocating Engine Back Pressure.	Theoretical B.T.U. per lb. of Steam.		
			R.E.	Turbine.	Total.
,ming 200 lbs. absolute at ciprocating engine to 28 . vacuum at condenser. lb. = .77 dryness	15	16	178	142	320
	$12\frac{1}{2}$	$13\frac{1}{2}$	189	131	320
	7	8	218	100	318

e Explanatory of Diagram (Fig. 4), showing Increase of Steam Energy Utilised by Combination of Reciprocating Engine and Turbine. (All Pressures given are Absolute.)

Quadruple Expansion Reciprocating Engine Exhausting to Condenser Direct.

ea enclosed by A, B, C, D, E = maximum energy realisable in quadruple engine, from 200 lbs. to 26 in. vacuum, with point of release at · 10 lbs. = 256 B.T.U.

Area cross-hatched represents energy which reciprocating engine cannot efficiently utilise, but which can be used in a turbine = 73 B.T.U.

Combination of Triple Expansion Reciprocating Engine Exhausting to Turbine and thence to Condenser.

Area enclosed by A, B, F, G, H = maximum energy realisable in triple engine from 200 lbs. to 8 lbs., with point of release at 13 lbs. = 219 B.T.U.

Area enclosed by J, K, L, M = energy available for turbine from 7 lbs. to 28 in. vacuum, receiving wet steam from reciprocating engine = 100 B.T.U.

Total energy of combination = 319 B.T.U.

Theoretically, total energy of combination is 24½ per cent. greater than that of quadruple engine.

It is estimated that a large portion of this additional energy can be expected to be realised by the combined system in the shape of increased power to drive the vessel, or, on the other hand, increased economy.

The theoretical figures are computed on the basis of adiabatic expansion throughout.

"In the case of a vessel which runs on service continually at or about her designed full speed, an initial pressure of about 7 lbs. absolute at the turbine appears most suitable. In a vessel which does part of her running at the designed power, and part at a considerably reduced power, it is desirable to design the turbines so that the initial pressure would not fall below 7 lbs. absolute when running under the lower conditions of power.

"It may be of interest at this stage to consider the disposition of the turbines in combination with reciprocating engines on board ship. The arrangement of the turbine, or turbines, depends greatly whether the vessel is to be fitted with single or twin-screw reciprocating engines. With a single reciprocating engine, one turbine, two turbines in 'series,' or two turbines in 'parallel' could be fitted, each turbine driving a separate shaft in addition to the reciprocator shaft. With twin-screw reciprocating engines, an arrangement of one turbine in the centre of the vessel, two turbines in 'parallel,' or two turbines in 'series,' could be adopted. The arrangement which seems to commend itself generally to shipowners and builders, where twin-screw reciprocating engines are fitted, is the arrangement with the turbine on the centre shaft.

"In the combination proposals set forth in columns B and C in the following table, it may be mentioned that in this particular inquiry the shipowners wished to have the advantage of the additional power and increase in speed of the vessel on the same coal consumption as for the twin quadruple engines. In some instances an increase in speed might not be desired, in which case the boilers and engines could be

the estimated amount of saving in consumption, so
ated horse-power of the combination did not exceed
twin quadruple engines. This would considerably
ight of machinery, and also the bunker capacity for

	A	B	C
	Twin Quadruple Reciprocating Engines.	Three-Cylinder Triple-Expansion Twin Reciprocating Engines, with Two Low-Pressure Turbines in Parallel.	Four-Cylinder Triple-Expansion Twin Reciprocating Engines, with One Low-Pressure Turbine.
:iprocating	25, 36½, 51½, 75	27, 42, 66	26, 39, 46, 46
- -	55	48	42
:iprocating - -	84	85	100
:iprocating - -	770	680	700
- -	213 lbs.	213 lbs.	213 lbs.
: at H.P. - -	200 lbs.	200 lbs.	200 lbs.
ines - -	...	7 lbs. absolute	7 lbs. absolute
ser - -	26 in.	28 in.	28 in.
on, main - -	95,000 lbs. per hour	95,000 lbs. per hour	95,000 lbs. per hour
; engines -	7,300	6,300	6,300
ent I.H.P. - -	...	2,000	2,000
- -	7,300	8,300	8,300
ower -	...	13.7 p. c.	13.7 p. c.
- -	15.5 knots	16.2 knots	16.2 knots
ı, lbs. per hour main - -	13 lbs.	11.45 lbs.	11.45 lbs.
of engines :iprocating - -	...	1,430 tons	1,455 tons
nstallation	...	65 "	70 "
ght - -	1,560 tons	1,495 "	1,525 "
ines - -	...	480	320

This saving in the weight of the machinery and in
enable the vessel to carry an equivalent addition
go. Then, again, if we take the indicated horse-
r the combination, and assume that quadruple

engines and boilers were required to give an equivalent power, the extra total weight of machinery would be, roughly, 160 tons, in addition to an increase of about 12 per cent. in coal consumption for the same power.

"Suitable arrangements are made for changing the flow of steam of the low-pressure cylinder exhaust of the reciprocating engine from the turbine to the condenser. This can be done in two or three ways. One method is to have an ordinary change valve of the piston type, or ordinary double-beat spring-loaded valve actuated by links connected to the weigh-shaft of the main engine, which would automatically change the flow of steam to the condenser when the engine was reversed. With this arrangement, when going ahead on one side of the ship, the steam from the reciprocating engine would flow through the turbine, but there does not appear to be any objection to this even if we consider the twin-screw reciprocating arrangement with a single turbine on the centre shaft. It might be rather an advantage, than otherwise, to allow the steam from the engine going ahead to pass through the turbine, as the centre propeller revolving would accelerate the feed of water on the rudder, and augment the turning power of the vessel.

"Another method would be to work these valves independent of the main engines, actuated by an hydraulic engine, or by an ordinary steam-driven reversing engine. With this arrangement, the low-pressure turbine would be cut out altogether and the reciprocating engine would exhaust to the condenser whether going ahead or astern during manœuvring."

Turbine Boilers, Condensers, and Auxiliary Machinery.

Most of turbine-driven ships in the merchant service have boilers of the usual Scotch type, but in the Navy the water-tube boiler has been largely adopted. These are usually of the Yarrow or Babcock & Wilcox type. The Scotch boilers are usually fitted with the Howden system of forced draught, but with the water-tube boiler the closed stokehold principle is in use. In the Navy liquid fuel has also been adopted—principally in high-speed coastal destroyers—and has given good results. Turbine condensers differ but little from the condenser of an ordinary reciprocating engine, but the connection for the exhaust pipe from the turbine is of extra large area, as owing to the low terminal pressure the volume of steam is excessive. Baffle plates, consisting of steel or bronze plates perforated with holes, are fitted in the interior of the condenser, and these tend to baffle the passage of the steam, and so use the cooling surface to the best extent. It is important in turbine installation to maintain a high vacuum, and to attain this result large air and circulating pumps are fitted. The circulating pumps are of the centrifugal type, and are driven by a separate engine. The air pumps are of the usual bucket type, and are also self-driven. In some ships "dry air"

e are sometimes driven by the main air

ry air pump is to draw the air only from the
wet air pump to deal with the water, thus
the condenser. The dry air pump is usually
turbines are running at full speed; whilst
pumps only are used.
the forced lubrication service, and there are
e acts as spare, and can be brought into use
overhauled. Feed and bilge pumps are also
; owing to the high running speed of the
en found practicable to drive any of the
them.

; IN TURBINE DESIGN

R JUNIOR STUDENTS).

ix, page 324, *for General Notes.*)

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am and Blade Speeds.

ice, the ratio of V_t to V_s varies from about .31

$_s$ = Ratio V_t, V_s
atio = V_t
atio = V_s
Blade Speed.
ed.

he required mean blade velocity if the ratio V_t to
d is to be 200 ft. per sec.
0 × .45 = 90 ft. per sec.

c. × 60 ÷ Revs. × 3.1416 = Diameter of rotor, across blades

blade velocity if the steam speed is 300 ft. per
4. Answer. 120 ft. per sec.

steam speed per sec. for a blade velocity of 90
V_s is equal to .45? Answer. 200 ft. per sec.

V_s is .48, and the steam speed 250 ft. per sec.
y. Answer. 120 ft. per sec.

350 ft. per sec., and the blade velocity 150 ft. per
$_t$ to V_s. Answer. .42 ratio.

Diameter of Rotors, &c.

RULE.—Rotor surface velocity per minute = Rotor diam. × 3.1416 × Revs.

Therefore, $\dfrac{\text{Velocity per min.}}{3.1416 \times \text{Revs.}} = \text{Diam. of rotor.}$

and, $\dfrac{\text{Velocity}}{\text{Diam.} \times 3.1416} = \text{Revolutions.}$

NOTE.—This rule neglects the difference in diameter due to the blade heights.

EXAMPLE. — Determine the required diameter of rotor, if the surface velocity is to be 110 ft. per sec. and the revolutions 250 per min.

$$\text{Diam. across blades} = \frac{\text{Blade velocity per min.}}{3.1416 \times \text{Revs.}} = \frac{110 \times 60}{3.1416 \times 250} = 8.4 \text{ ft.}$$

1. Calculate the required diameter of L.P. rotor for a velocity of 9,000 ft. per min., and revolutions of 400 per min. Answer. 7.16 ft. diameter.
2. Calculate the required diameter of H.P. rotor if the velocity is 6,000 ft. per min., and revolutions of 550 per min. Answer. 3.47 ft. diameter.
3. Calculate the required diameter of reverse rotor for a velocity of 5,000 ft. per min., and revolutions of 400 per min. Answer. 3.97 ft. diameter.
4. Determine the required revolutions for a turbine if the velocity is 8,000 ft. per min., and rotor diameter 6 ft. Answer. 424 revolutions.
5. Find the revolutions to correspond with a velocity of 9,000 ft. per min., and rotor diameter of 7 ft. Answer. 409 revolutions.
6. Calculate the necessary revolutions for a velocity of 100 ft. per sec., and rotor diameter of 5 ft. 6 in. Answer. 347 revolutions.
7. Find the revolutions for a velocity of 90 ft. per sec., and rotor diameter of 4 ft. Answer. 429 revolutions.
8. Find the revolutions required for a velocity of 160 ft. per sec., and rotor diameter of 14 ft. Answer. 218 revolutions

Steam Velocities and Heat Drops.

RULE.—British thermal units × 778 = Foot lbs. of energy given up.
and, Foot lbs. ÷ 778 = B.T.U. heat drop.

Again, $\dfrac{W \times V^2}{64.4} = \text{Foot lbs. of energy in first guide blades.}$

Where W = Weight in Pounds,
V = Velocity of Steam in feet per second.

Also, $\dfrac{(V^2 - v^2) \times W}{64.4} = \text{Foot lbs. given up in any moving blades.}$

Therefore, $64.4 \times \text{Foot lbs.} = W \times V^2$

and, $\sqrt{\dfrac{64.4 \times \text{Foot lbs.}}{W}} = V \text{ (Guide blades),}$

or, $\sqrt{\dfrac{64.4 \times \text{Foot lbs.} + v^2}{W}} = V \text{ (Moving blades).}$

locity of Steam in feet per second at exit edge of blades.
:locity of Steam in feet per second at admission edge of blades.
eight of Steam in pounds.
.2 × 2 (acceleration due to gravity per sec. per sec.).

—If the initial velocity of the steam is 220 ft. per sec. and
oo ft. per sec., calculate the kinetic energy given out in ft.
am supplied (neglecting losses). Also determine the heat

$$\frac{(V^2 - V^2) \times W}{64.4} = \frac{(300^2 - 220^2) \times 1}{64.4} = 645.9$$

Heat drop $= 645.9 \div 778 = .83$ B.T.U.

—The initial steam velocity is 130 ft. per sec., and the heat
eam supplied 1.8 units. Determine (1) the required exit
ne kinetic energy in ft. lbs. given out per lb. steam supplied
tage losses neglected).

$$V = \sqrt{\frac{64.4 \times \text{Foot lbs.} + V^2}{W}}$$

$$\frac{\times 1.8 \times 778 \times 130^2}{1} = \sqrt{\frac{64.4 \times 1400.4 + 16900}{1}}$$

$$\frac{35.76 + 16900}{1} = \sqrt{\frac{107085.76}{1}} = 327.5 \text{ ft. per sec.}$$

Kinetic energy $= 1.8 \times 778 = 1400.4$ ft. lbs.

ne work done in velocity acceleration by 1 lb. of steam in
ne guide blades of a Parsons turbine at an initial velocity of

Answer. 5,590 ft. lbs.

am enters the guide blades of a marine turbine at a velocity
Calculate (1) the work done and (2) the heat given up.

Answers. (1) 1,397 ft. lbs. (2) 1.79 B.T.U.

) the ft. lbs. of kinetic energy done in 1 lb. of steam when
guide blades of a turbine at a velocity of 350 ft. per sec.;
r.

Answers. (1) 1,902 ft. lbs. (2) 2.44 B.T.U.

1) the ft. lbs. of energy and (2) drop of heat which are
of steam at a velocity of 15,000 ft. per min.

Answers. (1) 970 ft. lbs. (2) 1.24 B.T.U.

steam enters the moving blades of a marine turbine at a
. per sec., and leaves at a velocity of 300 ft. per sec. Cal-
bs. of energy given out and (2) the heat drop.

Answers. (1) 776 ft. lbs. (2) .99 B.T.U.

6. Find the heat drop if the steam velocities are 300 ft. and 40
Answer.

7. Find the required heat drop to give velocities of 300 ft. per sec. in the moving blades. Answer.

8. Calculate the steam velocity per sec. if the energy given out lbs. for each lb. of steam supplied. Answer. 35

9. Find what velocity of steam is necessary to develop 1,00 energy per lb. of steam. Answer. 25

10. Twenty lbs. of steam passing through the guide blades give out 10,000 ft. lbs. of work. Calculate the velocity of the stea
Answer. 179

11. One lb. of steam drops 2 B.T.U. in passing through the turbine. Calculate the steam velocity. Answer. 31

12. What steam velocity is required to obtain a heat drop of per lb. of steam? Answer. 35

13. Calculate the exit velocity of the steam in the turbine b ft. lbs. given out per lb. are 1,500 and the initial velocity is 200 ft.
Answer. 369

14. The initial velocity is 300 ft. per sec., and the heat units Calculate the steam velocity at the exhaust edge. Answer 436

15. What exit velocity is necessary if the heat drop is 1.5 B.T. initial velocity 260 ft. per sec.? Answer. 37

16. Heat drop 3 B.T.U., initial velocity 6,000 ft. per min. C exit velocity. Answer. 40

17. One lb. of steam contains 1,200 B.T.U. on entering the 1,196 B.T.U. on leaving the blades of a turbine. Calculate the if the admission velocity is 300 ft. per sec. Answer. 53

18. Calculate the exit velocity of the steam if the initial veloci per sec., and the heat drop 1.75 B.T.U. Answer. 33

19. Initial velocity of steam 200 ft. per sec., and heat drop Calculate the exit velocity. Answer. 339

20. In a marine turbine 1 lb. of steam at an initial velocity of sec. passes through the moving blades. Find the final velocity drop is 1.8 B.T.U. Answer. 36

Blade Velocities and Number of Rows.

Rule.—Velocity per sec.^{2} × Number of Rows = Constant.

Therefore, Constant ÷ Velocity^2 = Number of Rows,

and, $\sqrt{\text{Constant} \div \text{Number of Rows}}$ = Velocity.

Note.—The Constant referred to varies from 1,400,000 to 1,600,000 (see

Example.—Determine the approximate total number of blad set of turbines (1 H.P. and 2 L.P.) if the surface velocity of the H

xceed 95 ft. and the L.P. drums 125 ft. per sec.; the Constant adopted 000.

H.P.

en, $95^2 \times \text{No.} = 1{,}500{,}000.$

ore, $1{,}500{,}000 \div 95^2 = 166$ rows, assuming total power in H.P. turbine.
ien, $160 \times \frac{1}{3} = 53$ rows, assuming $\frac{1}{3}$ power in H.P. turbine.

L.P.

$125^2 \times \text{No.} = 1{,}500{,}000.$

re, $1{,}500{,}000 \div 125^2 = 96$ rows, assuming total power in L.P. turbines.
:n, $96 \times \frac{2}{3} = 64$ rows, assuming $\frac{2}{3}$ power in L.P. turbines.

alculate the required total number of rows of ahead blades required arsons marine turbine if the constant is 1,400,000 and the blade 100 ft. per sec. Answer. 140 rows of blades.

etermine the total number of ahead blade rows required for the three f a turbine if the blade velocity is to be 90 ft. per sec. and the allowed is 1,500,000. Answer. 185 rows of blades.

ind the suitable mean blade velocity if the number of rows is 192 constant 1,600,000. Answer. 91.2 ft. per sec.

/hat blade velocity per sec. will be necessary if the ahead turbine ontain in all 168 rows of blades and the given constant is 1,512,000? Answer. 94.8 ft. per sec.

'he ahead H.P. and L.P. (2) turbine rotors each contain 48 rows of Calculate the required mean blade velocity if the constant is 10. Answer. 102 ft. per sec.

etermine the number of blade rows required in each ahead rotor if n blade velocity is 120 ft. per sec. and the constant 1,600,000. Answer. 111 rows in all; 37 rows in each rotor.

etermine the constant used when the mean blade velocity is 105 ft. and the number of rows 144. Answer. 1,587,600 constant.

'he number of ahead blade rows is 156 and the mean blade velocity er sec. Calculate the constant which has been employed. Answer. 1,560,000 constant.

etermine the number of blade rows required in the H.P. turbine and L.P. turbine, given that the H.P. rotor diameter is 42 in., the L.P. ameter 60 in., the revolutions per min. 500, and the constant 10.

Answers. { H.P. turbine, 65 rows.
L.P. turbines (each), 63 rows.

Calculate the number of blade rows required in the H.P. and L.P. of cruisers of the "Indomitable" and "Inflexible" class, given that . rotor is 92 in. diameter, the L.P. rotor 115 in. diameter, the revolu- esigned) 260 per min., and the constant adopted 1,300,000.

—2 H.P. and 2 L.P. ahead turbines.

Answers. { H.P. turbines, 60 rows.
L.P. " 38 "

Turbine Propeller Calculations.

RULE.— $\sqrt{\text{Effective thrust on shaft}} = C \times$ Diameter of

Therefore, $\frac{\sqrt{\text{Effective thrust}}}{C} =$ Diameter of pro

And, $\frac{\sqrt{\text{Effective thrust}}}{\text{Diameter}} = C.$

NOTE.—C = Constant found from suitable coefficient for blade are pressure per square inch.

EXAMPLE.—Calculate the required diameter of each "Lusitania," given that the effective thrust on each shaft is the Constant is 23.

Then, Diam. of propeller $= \frac{\sqrt{\text{Thrust lbs.}}}{C} = \frac{\sqrt{121460}}{23} =$ 1

1. Determine the diameter of turbine propeller suitable thrust on each shaft is 23,180 lbs. and the constant 29.

Answer. 5

2. Calculate the diameter of turbine propeller required per shaft is 24,000 lbs. and the constant C 30. Answer. 5

3. Find the required diameter of propeller if the eff 60,000 lbs. on each shaft and the constant 24.63.

Answer. 9

4. Calculate the propeller diameter if the effective thrust found to be 44,000 lbs. and the constant 29. Answer. 7

5. What diameter of propeller should be employed wh 30,000 lbs. on each shaft and the constant is 30?

Answer. 5

6. Determine the effective thrust pounds if the propeller and the constant 31. Answer. 24,025 lbs. thru

7. What is the effective thrust in lbs. if the propeller di and the constant 21? Answer. 86,436 lbs. thru

8. What diameter of propeller should be used when the per shaft is 25,000 lbs. and the constant 28? Answer. 5

9. What diameter of propeller should be fitted to each when the effective thrust is 91,476 lbs. and the constant is 21

Answer. 1

10. What diameter of propeller should be used when the is 188,051 lbs. and the constant 25, the steamer having 4 len and 4 propellers? Answer. 1

Pressure Drops.

RULE.—

$\frac{\text{Absolute initial pressure} - \text{Absolute final pressure}}{\text{Number of blade rows}} =$ Mean

or, Drop per row × Number of rows + Final pressure = Ini

NOTE.—Take atmospheric pressure as 15 lbs. per square inch.

EXAMPLE.—Determine the mean pressure drop per blade and L.P. turbines, given the following :—H.P. initial pressure,

H.P. terminal pressure, 16 lbs. gauge; L.P. initial pressure, 15 lbs. gauge; L.P. initial pressure, 27 in. vacuum. Each turbine contains 60 rows of blades.

Then, $145 + 15 = 160$ lbs. absolute initial pressure.
And, $16 + 15 = 31$ „ terminal „

Therefore, Mean pressure drop in H.P. turbine $= \frac{160 - 31}{60} = 2.15$ lbs.

Again, $15 + 15 = 30$ lbs. absolute initial pressure.

And, $\left(\frac{30 \text{ in.} - 27 \text{ in.}}{2}\right) = 1.5$ „ terminal „

Therefore, Mean pressure drop in L.P. turbines $= \frac{30 - 1.5}{60} = .475$ lb.

NOTE.—Barometer height = 30 in. at ordinary atmospheric pressure.

1. Calculate the mean pressure drop per row if the H.P. turbine initial gauge pressure is 140 lbs. and the L.P. turbine initial gauge pressure is 20 lbs., the rotor containing 48 rows of blades Answer. 2.5 lbs. drop per row.

2. The H.P. rotor consists of 4 expansions, each containing 16 rows of blades. Calculate the mean pressure drop per each expansion and per each row if the initial pressure is 150 lbs. gauge and the final pressure 22 lbs. gauge. Answer. 32 lbs. drop per expansion; 2 lbs. drop per row.

3. The L.P. turbine gauges show 20 lbs. pressure and the condenser vacuum gauge 27 in. Calculate the mean pressure drop per expansion and per row if the rotor consists of 8 expansions, each containing 8 rows of blades.
Answer. 4.18 lbs. drop per expansion; .523 lb. drop per row.

4. In running "full astern" the reverse turbine gauges show 80 lbs. pressure and the condenser gauges 27 in. vacuum. Calculate the pressure drop per expansion and per blade row if the L.P. rotors are made up of 8 expansions, each containing 7 rows of blades.
Answer. 11.68 lbs. drop per expansion; 1.66 lbs. drop per row.

5. In running "half speed ahead" the H.P. turbine gauge indicates 40 lbs., and the L.P. turbine gauges each 12 in. vacuum with a condenser vacuum of 28 in. Calculate the pressure drop per blade row if the H.P. and L.P. rotors each contain 56 rows of blades.

Answers. $\begin{cases} \text{.82 lb. drop per row in H.P. turbines.} \\ \text{.142 „ „ L.P. „} \end{cases}$

6. In running "slow ahead" the H.P. turbine gauge shows 10 lbs. pressure and the L.P. turbine gauges each 21 in. vacuum, the condenser vacuum being 28 in. Calculate the pressure drop per expansion and per blade row if the H.P. contains 4 expansions of 16 rows each, and the L.P.'s contain 8 expansions of 8 rows each.

Answers. $\begin{cases} \text{5.125 lbs. drop per expansion, H.P. turbines.} \\ \text{.32 lb. drop per row, H.P. turbines.} \\ \text{.437 lb. drop per expansion, L.P. turbines.} \\ \text{.0546 lb. drop per row, L.P. turbines.} \end{cases}$

7. Calculate the required initial H.P. pressure by gauge, if the L.P. initial pressure is 16 lbs. gauge and the mean pressure drop per row is to be 2 lbs. The H.P. rotor consists of 4 expansions, each containing 16 rows of blades.
Answer. 144 lbs. gauge pressure.

in, $259.3 + 461 = 720.3$ absolute Temperature.
re,
t drop $= 859.6 \times .99 - 931.6 \times .95 + 822.1 - 720.3$
$= 851.004 - 885.020 + 822.1 - 720.3$
$(851.004 + 822.1 - 885.02 + 720.3)$
$= 1673.104 - 1605.32 = 67.78$ B.T.U. per lb.
in, $67.78 \times 778 \times .65 = 34876.3$ Foot lbs. given up.
n, $90000 \div 60 = 1500$ lbs. Flow per minute.

$$\text{Horse-power} = \frac{1500 \times 34876.3}{33000} = 1585$$

consump. per horse-power per hour $= 90000 \div (1585 \times 3) = 18.9$ lbs.

Calculate the heat drop per lb. of steam occurring during one H.P. expansion, if the pressure falls from 135 lbs. gauge to 125 lbs. gauge, ness factor being 1 before expansion and .996 after expansion.

Answer. 4.96 B.T.U. given up.

Calculate the heat drop per lb. of steam throughout one H.P. ex- if the pressure falls from 160 lbs. absolute to 130 lbs. absolute, the factors being 1 and .98. Answer. 21.87 B.T.U. given up.

Calculate the total heat drop per lb. of steam throughout the H.P. if the initial pressure by gauge is 135 lbs. and the final pressure by s 12 lbs., the dryness factors being 1 and .9.

Answer. 127.42 B.T.U. given up.

Calculate the total heat drop per lb. of steam which occurs in the L.P. if the initial pressure is 12 lbs. gauge and terminal pressure 27 in. , the dryness factor being .9 and .78.

Answer. 170.58 B.T.U. given up.

Calculate the heat drop per lb. of steam which takes place (1) in the rbine, and (2) in each L.P. turbine, assuming that the L.P. turbines ceive only half the weight of steam used by the H.P. turbine, given e H.P. initial pressure is 140 lbs. gauge, the L.P. turbine initial e 16 lbs. gauge, and the condenser vacuum 28 in.; the dryness factors med to be .95, .9, and .76.

Answers. (1) 82.49 B.T.U. in H.P. turbine.
(2) 100.263 ,, each L.P. turbine.

Taking the efficiency as 56 per cent., calculate (1) the effective work er lb. of steam in the previous question, and (2) the horse-power ed if 60,000 lbs. of steam are passing through the turbines per hour; the steam consumption per horse-power per hour.

Answers. (1) 123304.4108 ft. lbs.
(2) 3736.49 horse-power.
(3) 16.05 lbs. steam per hour per H.P.

Calculate the B.T.U. given up and transformed into kinetic energy per steam, also the combined horse-power of the turbines, for the g:—

H.P. turbine initial pressure, 150 lbs. gauge.
L.P. turbine exhaust pressure, 26 in. vacuum.
Dryness factors, .99 and .76 respectively.
Turbine efficiency, 54 per cent.
Steam consumption, 90,000 lbs. per hour.

Answers. 129283.52 ft. lbs. kinetic energy per lb. steam.
5876.5 horse-power.

er gauges are fitted to the casings
egistered on the H.P. were as

itial), 140 lbs. (gauge).
" 100 " "
" 65 " "
" 40 " "
ıinal), 20 " "

.86, .84, and .82 respectively, and
eam which take place at each
ıt this turbine, neglecting friction

25.94	B.T.U.	1st	expansion.
27.69	"	2nd	"
27.91	"	3rd	"
29.96	"	4th	"
111.5	B.T.U. total heat drop.		

' gauges were fitted to the casings
ːred on the H.P. turbine were as

h expansion (initial), 40 lbs.
" (terminal), 20 "

, .96, .945, and .92, and calculate
place at each expansion, and the
neglecting frictional losses.

24.31	B.T.U.	1st	expansion.
25.85	"	2nd	"
21.58	"	3rd	"
32.56	"	4th	"
104.30	B.T.U. total heat drop.		

tted on the H.P. turbine casing
. at each expansion, and the H.P.
the dryness factors to be .9, .88,
e theoretical heat drop at each
the horse-power developed in the
o per cent., and the steam con-

25.286	B.T.U.	1st	expansion.
18.8183	"	2nd	"
28.810	"	3rd	"
20.64	"	4th	"
93.55	total heat drop.		
1801.1	horse-power.		

'steam per hour, and the pressures
ı were 1

tors being .836
) the heat drop
(3) the effectiv
ency being 76

nswers.

I

12

d," the H.P.
23 lbs. Assu
te the heat dr
H.P. turbine.

Answers.

I

question, if
s., 14 lbs., 8 lb
l to 28 in. vac
75, .86, .84, .8
the total heat
the turbine eff
. of coal, calcul
per hour, (3) c
ed per day of 2

18.999	B.T
18.892	"
19.2775	"
19.3395	"
24.08	"
35.651	"
40.765	"
45.845	"
222.8490	B.

(1) 90,360 lb
(2) 13.9 lbs.
(3) 1.61 lbs.
(4) 112 1 to

with the two L.P. reverse turbines, the gauge
nitial end, and 26 in. vacuum at the exhaust
eat drop in each L.P. turbine per lb. of steam
rs of .97 and .76, (2) the horse-power obtained
m consumption is 1,340 lbs. per min., and (3)
er per hour if the efficiency of the turbines is

) 227.418 B.T.U. total heat drop.
) 2,541 horse-power in each L.P.
) 15.8 lbs. of steam per horse-power per hour.

e is 145 lbs. gauge and the L.P. terminal pres-
te (1) the mean turbine efficiency if the horse-
otal steam consumption 2,000 lbs. per minute.
nsumption per horse-power per hour assuming

(1) .54 mean turbine efficiency.
(2) 15 lbs. steam per horse-power per hour.

in a turbine channel steamer as measured
ion meter worked out as 6,500 and the coal
l per hour. Calculate the turbine efficiency if
gauge and the L.P. turbine terminal pressure
e dryness factors of .98 and .75. Evaporation
Answer. Turbine efficiency .53.

ssure 140 lbs. gauge, H.P. turbine terminal
rbine initial pressure 20 lbs. gauge, and the
suming dryness factors to be .99, .91, .9, .75,
hours' steaming, evaporation 8.5 lbs. of water
-power by "Denny-Johnson" torsion meter
theoretical heat drop per lb. of steam, and (2)
(3) the consumption of steam, and (4) coal
ean turbine efficiency.

1) 317.914 B.T.U. total theoretical heat drop.
2) 247337.09 ft. lbs.
3) 13.7 lbs. steam per H.P. hour.
4) 1.61 lbs. coal per H.P. hour.
5) 58 per cent. mean turbine efficiency.

B.T.U. given up in each turbine per lb. of
bine, the horse-power developed in each, and
hour, given the following:—
pressure 140 lbs. gauge.
" 20 " "
al " $1\frac{1}{2}$ " absolute.
n the dryness factors are 1, .93, and .78. The
f steam per hour, and the over-all efficiency is

57 B.T.U. in H.P. turbine.
61.16 B.T.U. in each L.P. turbine.
2508.3 horse-power in H.P. turbine.
2691.6 horse-power in each L.P. turbine.
14.1 lbs. steam per horse-power per hour.

; and Blade Heights.

$\frac{\times \text{Volume}}{} =$ Cub. ft. flow per sec.

relocity per sec.

ule given (page 55).

ed clear area between blades.

it area × Annulus factor =

Diameter across blade tips.

$\frac{\text{m diameter}}{} =$ Required blade heights.

bout 3 for H.P. turbine blades to 2 or less for
e blade thickness and exit angle, which angle
e 1st H.P. expansion to about 65 degs. in the

r 4 places).
xpansions. Blade height = 1st expan-

1.41 or 1.41, but in special designs may only

Expansions.

$\frac{\text{ratio}}{\text{'. turbines}} =$ 1st L.P. blade height.

s 6,500, the steam consumption 13 lbs.
400, and the initial pressure 160 lbs.
hts at the 1st H.P. expansion, if the
eter is to be 3 ft. 9 in.; also calculate
t expansion, and (3) the ratio V_b V_r.

$\frac{0 \times 13 \times 2.56}{60 \times 60} = 60.08$ cub. ft.

$\frac{0}{50} = 220$ ft. per sec.

$8 \div 220 = .273$ sq. ft.

$\frac{3.75^2 \times .7854 + .273 \times 3}{.7854} = 3.88$ ft.

$\frac{86 - 3.75) \times 12}{2} = .816$, say $\frac{7}{8}$ in.

$+ (.875 \times 2) = 46\frac{3}{4}$ in.

$\left.\frac{1416 \times 400}{60}\right) \div 220 = .35.$

oo, revolutions 500 per minute, and the
-power hour; the H.P. turbine initial
and the corresponding steam volume
area factor is 3, and the rotor drum
the required blade heights, and (2) the
. the 1st expansion (neglecting blade tip
and N = 144 rows.

(1) $1\frac{3}{8}$ in. blade height at 1st expansion.
(2) $50\frac{3}{4}$ in. diameter of casing (inside).

ghts at the 2nd, 3rd, and 4th expansions
being 1.4.

nd expansion blades,	1.92 in.,	say	2	in.
d „	2.80	„	$2\frac{3}{4}$	„
h „	3.85	„	$3\frac{7}{8}$	„

same turbine set is 68 in. diameter,
2nd, 3rd, 4th, 5th, and 6th expansions,
ng" blades.

t expansion blades,	1.38 in.,	say	$1\frac{3}{8}$	in.
nd „	1.92	„	2	„
d „	2.80	„	$2\frac{3}{4}$	„
h „	3.85	„	$3\frac{7}{8}$	„
h „	5.42	„	$5\frac{1}{2}$	„
h „	7.70	„	$7\frac{3}{4}$	„
h „	"wing,"		$7\frac{3}{4}$	„
h „	„		$7\frac{3}{4}$	„

oo, the revolutions 450 per minute, the
-power hour, the H.P. turbine initial
9 cub. ft. per lb.), the annular factor is
is 3 ft. 6 in. Determine the required
on if the velocity constant is 2,700 and
Answer. $\frac{7}{8}$ in. blades at 1st expansion.
ghts of the 2nd, 3rd, and 4th expansions
atio being 1.41.

nd expansion blades,	1.22 in.,	say	$1\frac{1}{4}$	in.
rd „			$1\frac{3}{4}$	„
th „	2.45	„	$2\frac{1}{2}$	„

ights of the L.P. expansions, the drum
and 8th expansions being fitted with
ed annulus factor, assuming the steam
ectively 350 ft. and 380 ft. per sec., and
cub. ft. per lb.

1st expansion blades,	.86 in.,	say	$\frac{7}{8}$	in.
2nd „	1.22	„	$1\frac{1}{4}$	„
3rd „			$1\frac{3}{4}$	„
4th „	2.45	„	$2\frac{1}{2}$	„
5th „			$3\frac{1}{2}$	„
6th „	4.9	„	5	„
7th „			5	„

m 4 ft. 6 in., the blade height ratio 1.42,)lades).

st expansion blades, 1.56 in., say $1\frac{1}{2}$ in.
nd " 2.13 " $2\frac{1}{8}$ "
rd " 3.19 " $3\frac{1}{4}$ "
th " 4.61 " $4\frac{5}{8}$ "

e, the drums being 6 ft. 6 in. diameter h "wing" blades of more than normal of the 6th expansion.

st expansion blades, 1.58, in., say $1\frac{5}{8}$ in.
nd " 2.30 " $2\frac{3}{8}$ "
rd " 3.37 " $3\frac{3}{8}$ "
th " 4.79 " $4\frac{3}{4}$ "
th " 6.74 " $6\frac{3}{4}$ "
th " 9.58 " $9\frac{1}{2}$ "
th " $9\frac{1}{2}$ "
th " $9\frac{1}{2}$ "

required for the H.P. turbine of four f the turbine horse-power is 5,500, the orse-power hour, the initial pressure 180 l blade rows N 121. The rotor drum is ight ratio 1.4.

pansion blade heights, .54 in., say $\frac{1}{2}$ in.
" " .7 " $\frac{3}{4}$ "
" " 1.05 " 1 "
" " 1.4 " $1\frac{1}{2}$ "

L.P. turbine blade heights (eight expan- in. diameter. The last two expansions th expansion.

pansion blade heights, .46 in.; say $\frac{1}{2}$ in.
" " .70 " $\frac{3}{4}$ "
" " 1.05 " 1 "
" " 1.4 " $1\frac{1}{2}$ "
" " 2.1 " $2\frac{1}{8}$ "
" " 2.97 " 3 "
" " 3 "
" " 3 "

'. turbine blade heights for a cruiser of vo L.P. turbines being fitted for ahead ave six expansions, the blade height ratio 5 ft. 6 in., the estimated steam consump- he initial pressure 185 lbs., the velocity rows N 144.

pansion blade heights, .78 in., say $\frac{7}{8}$ in.
" " 1.22 " $1\frac{1}{4}$ "
" " $1\frac{3}{4}$ "
" " 2.45 " $2\frac{1}{2}$ "
" " $3\frac{1}{2}$ "
" " 4.9 " 5 "

:onstant
150 and

ıred by
ft., the
s shows
P. 600,
iameter.

e-power.
"
"

e-power.

P.

ıaft at a
and the
he shaft
30, and
he total

e-power.
"
"

e-power.

P.

= 25 ft.,
P.L.P.,

e-power.
"
"

e-power.

P.

afts are
nce L is

The four lengths of shafting are each 22 in. in the total equivalent I.H.P. (approximate).

Answers.	S.H.P.	Turbine	16971	horse-power.
	S.L.P.	"	16971	"
	P.H.P.	"	16971	"
	P.L.P.	"	16971	"
			Total 67884	horse-power.
			75426 I.H.P.	

ver Developed in Blade Rows.

$$\frac{{}^2_{s_2}) \times \text{Blade efficiency} \times \text{Lbs. steam per minute}}{64.4 \times 33000}$$

= Horse-power per row of blades.

w × Number of rows per expansion

= Horse-power developed in one " expansion."

1 Exit Velocity per second.

n Admission Velocity per second.

× 2 Gravity's Acceleration per second per second.)

lbs. per minute per horse-power.

$$\left(\frac{\text{in ft.} \times 3.1416 \times \text{Revs.}}{60}\right) \div V_{s_2} = \text{Ratio } V_t \text{ to } V_{s_2}.$$

(1) the horse-power developed in one blade row, on of 10 rows, and (3) the total effective heat drop on, given the following:—Steam initial velocity 160 250 ft. per sec., blade efficiency 85 per cent., and minute. Also express the ratio V_t, V_s at this expanmeter is 4 ft. and the revolutions are 480 per min.

$$\text{row} = \frac{(250^2 - 160^2) \times 2400 \times .85}{64.4 \times 33000} = 35.4 \text{ horse-power.}$$

per expansion = 35.4 × 10 = 354 horse-power.

÷ 778 = 15015 B.T.U.

orse-power developed at one row of blades of the turbine, given that the steam flow is 22.5 lbs. per ency .847. The steam admission velocity is 217 ft. city 322 ft. per sec. Also calculate (2) the horse- 1st expansion, which contains 8 rows of similar ctive heat drop in B.T.U. If the mean blade d the revolutions 650 per minute, also (4) express is expansion.

Answers.
- (1) 30.45 horse-power per blade row.
- (2) 243.6 horse-power per expansion.
- (3) 10332.6 B.T.U. heat drop.
- (4) .39 ratio V_t to V_{s_2}.

2. Calculate (1) the horse-power given out at one row of blades of the 6th expansion of the same H.P. turbine, given that the initial velocity of the steam is now 245 ft. per sec., the exit velocity 360 ft. per sec., and blade efficiency .83. Also (2) find the horse-power of this expansion, which contains 8 rows of blades, and (3) the effective heat drop. Express (4) the ratio of V_t to V_{s3} if the mean blade diameter is 3 ft. $9\frac{1}{4}$ in.

Answers.
- (1) 36.68 horse-power per blade row.
- (2) 293.44 horse-power per expansion.
- (3) 12446.6 B.T.U. heat drop.
- (4) .356 ratio V_t to V_{s3}.

3. Referring to question No. 1, calculate (1) the horse-power developed at one row of blades of the L.P. 1st expansion, given that each L.P. turbine receives only half the steam supplied to the H.P. turbine. The steam initial velocity is 350 ft. per sec., the exit velocity 487 ft. per sec., and blade efficiency .82. Also (2) find the horse-power given out by one expansion of 8 blade rows, and (3) the corresponding heat drop in B.T.U. Finally (4) express the ratio of V_t to V_{s3} if the mean diameter across the blades is 4 ft. $5\frac{1}{4}$ in.

Answers.
- (1) 29.86 horse-power per blade row.
- (2) 238.88 horse-power per expansion.
- (3) 10132.44 B.T.U. heat drop.
- (4) .310 ratio V_t to V_{s3}.

4. Find (1) the horse-power developed between one row of blades of the 4th L.P. expansion, given that the steam flow is as before stated, and the blade efficiency .76. The steam admission velocity is 397 ft. per sec., and the exit velocity 541 ft. per sec. Also (2) find the horse-power of this expansion of 8 blade rows, and (3) the effective heat drop. Finally (4) determine the ratio V_t to V_{s3} if the mean blade diameter is 4 ft. $7\frac{1}{2}$ in.

Answers.
- (1) 32.6 horse-power per blade row.
- (2) 260.8 horse-power per expansion.
- (3) 11062.2 B.T.U. heat drop.
- (4) .29 ratio V_t to V_{s3}.

5. Repeat the foregoing for the 5th L.P. expansion, if $V_{s2} = 475$ ft., $V_{s3} = 600$ ft., blade efficiency .74, and mean blade diameter $= 4$ ft. 8 in. Eight rows of blades.

Answers.
- (1) 31.58 horse-power per blade row.
- (2) 252.64 horse-power per expansion.
- (3) 107160.9 B.T.U. heat drop.
- (4) .264 ratio V_t to V_{s3}.

6. Repeat the foregoing for the 6th L.P. expansion, if $V_{s2} = 646$ ft., $V_{s3} = 750$ ft., blade efficiency .715, and mean blade diameter $= 4$ ft. 9 in. Eight rows of blades.

Answers.
- (1) 32.97 horse-power per blade row.
- (2) 263.76 horse-power per expansion.
- (3) 11187.7 B.T.U. heat drop.
- (4) .215 ratio V_t to V_{s3}.

Condensed in Turbines.

– dryness factor) = water produced by nature of expansion.

ınt of water drained off from the L.P. turbines by when the steam flow per hour is 112,000 lbs. ıe steam at the last L.P. expansion is .85.

Answer, 16,800 lbs.

6,000 and the steam consumption is $13\frac{1}{2}$ lbs. per ne the pounds of water condensed in the turbines he last L.P. expansion is 3 lbs. absolute.

factor from table of "Actual Steam Volumes,"

Answer, 12,312 lbs.

ıt of water drained off per hour from each L.P. ıle," given that the horse-power at 27 knots was ption 13 lbs. per horse-power per hour. Assume xpansion of .835.

Answer, 21.54 tons.

10,000 and the steam consumption 14 lbs. per ulate the weight of steam condensed in the H.P. essure at the last H.P. expansion is 20 lbs. (gauge) , assuming the initial steam to be dry.

Answer, 158.6 lbs. per minute.

ıs question, and estimate the water condensed in minute if the dryness factor at the last L.P.

Answer, 350 lbs. per minute.

water drained off by the "wet" air pumps from Lusitania" per hour if the horse-power is 68,000, team per horse-power per hour. Dryness factor,

Answer, 63.2 tons.

ONS TO PROBLEMS.

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am and Blade Speeds.

.4 = 120 ft. per sec. Answer.

45 = 200 ft. per sec. Answer.

.48 = 120 ft. per sec. Answer.

350 = .42. Ratio. Answer.

The Marine Steam Turbine.

Diameter of Rotors, &c.

1. $\text{Diameter} = \dfrac{9000}{3.1416 \times 400} = 7.16$ ft. Answer.

2. $\text{Diameter} = \dfrac{6000}{3.1416 \times 550} = 3.47$ ft. Answer.

3. $\text{Diameter} = \dfrac{5000}{3.1416 \times 400} = 3.97$ ft. Answer.

4. $\text{Revolutions} = \dfrac{8000}{6 \times 3.1416} = 424.$ Answer.

5 $\text{Revolutions} = \dfrac{9000}{7 \times 3.1416} = 409.$ Answer.

6. $\text{Revolutions} = \dfrac{100 \times 60}{3.1416 \times 5.5} = 347.$ Answer.

7 $\text{Revolutions} = \dfrac{90 \times 60}{3.1416 \times 4} = 429.$ Answer.

8. $\text{Revolutions} = \dfrac{160 \times 60}{3.1416 \times 14} = 218.$ Answer.

Steam Velocities and Heat Drops.

1. $\dfrac{1 \times 600^2}{64.4} = 5590$ ft. lbs. Answer.

2. $\dfrac{1 \times 300^2}{64.4} = 1397$ ft. lbs.

$1397 \div 778 = 1.79$ B.T.U. Answer.

3. $\dfrac{1 \times 350^2}{64.4} = 1902$ ft. lbs.

$1902 \div 778 = 2.44$ B.T.U. Answer.

4. $\dfrac{1 \times (\frac{15000}{60})^2}{64.4} = 970$ ft. lbs.

$970 \div 778 = 1.24$ B.T.U. Answer.

5 $\dfrac{(300^2 - 200^2) \times 1}{64.4} = 776$ ft. lbs.

$776 \div 778 = .99$ B.T.U. Answer

6. $\dfrac{(400^2 - 300^2) \times 1}{64.4} = 1086.$

$1086 \div 778 = 1.39$ B.T.U. Answer.

7 $\dfrac{(300^2 - 220^2) \times 1}{64.4} = 645.$

$\therefore\ 645 \div 778 = .82$ B.T.U. Answer.

Propeller Calculations.

meter. Answer.

meter. Answer.

meter. Answer.

meter. Answer.

meter. Answer.

thrust. Answer.

thrust. Answer.

meter. Answer.

meter. Answer.

ameter. Answer.

'ressure Drops.

$\frac{+15)}{}$ = 2.5 lbs. per row. Answer.

$\frac{+15)}{}$ = 2 lbs. per row. Answer.

$\frac{+15)}{}$ = 32 lbs. per expansion. Answer.

1.5 lbs. absolute.

4.18 lbs. per expansion. Answer.

.523 lb. per row. Answer.

= 1.5 lbs. absolute.

1.66 lbs. per row. Answer.

11.68 lbs. per expansion. Answer.

team Turbine.

osolute.
solute.

row in H.P. Ans

L.P. Answer.

absolute.
solute.

. per expansion in

er row in H.P. A

nsion in L.P. An

L.P. Answer.

9 lbs. absolute.
Answer.

absolute.
os. absolute.
uge. Answer.

os. absolute.
lbs. absolute.
bs. gauge on L.P.
.79 lbs. absolute.
45 lbs. gauge on H

Calculations.

5 + 15 = 140 lbs.
9 + 461 = 813.9.
19.3 − 813.9 = 4.96

15 = 145 lbs.
5 + 461 = 816.6°.
1.8 − 816.6 = 21.87

5 = 27 lbs.
4 + 461 = 705.4°.
.3° − 705.4° = 127.4

4. $12+15=27$ lbs.; 27 in. vacuum $=1.5$ lbs. absolute.
$244.4+461=705.4$; $115.9+461=576.9$.
$\therefore$ $942.2\times.9-1033.2\times.78+705.4-576.9=170.58$ B.T.U. Answer.

5. $140+15=155$ lbs.; $16+15=31$ lbs.
$361+461=822.1^\circ$; $252+461=713.2$.
$\therefore$ $859.6\times.95-936.7\times.9+822.1-713.2=82.49$ B.T.U. in H.P. Answer.

NOTE.—28 in. vacuum $=1$ lb. absolute pressure.
$102.1+461=563.1^\circ$.

Then, $936.71\times.9-1042.9\times.76+713.2-563.1=\dfrac{200.52}{2}$
$=100.263$ B.T.U. in L.P. Answer.

6. Total B.T.U.'s $=82.49$ for H.P., and 200.526 for two L.P.'s $=283.016$ B.T.U.
$\therefore$ $283.016\times778\times.56=123304.4108$ ft. lbs.

And $\dfrac{123304.4108\times60000}{33000\times60}=3736.49$ horse-power.

Then, $\dfrac{60000}{3736.49}=16.05$ lbs. steam per hour per horse-power. Answer.

7. $150+15=165$ lbs.; 26 in. vacuum $=2$ lbs. absolute.
$366+461=827$; $126.3+461=587.3$.
$\therefore$ $856.2\times.99-1025.8\times.76+827-587.3=307.73$ B.T.U.
And $307.73\times778\times.54=129283.5276$ ft. lbs. kinetic energy. Answer.

Again, $\dfrac{129283.5276\times90000}{33000\times60}=5876.52$ horse-power. Answer.

8. $145+15=160$ lbs.; 25 in. vacuum $=2\frac{1}{2}$ lbs.
$363.6+461=824.6$; $134.6+461=595.6$.
$\therefore$ $857.8\times.98-1019.9\times.78+824.6-595.6=153.508$ B.T.U.'s.
And $33000\times60\div153.508\times778=16.57$ lbs. steam per hour per H.P.,
and $16.57\div8.6=1.92$ lbs. coal per hour per H.P. Answer.

9. $20+15=35$ lbs.; 26 in. vacuum $=2$ lbs.
$259+461=720.3$; $126.3+461=587.3$.
$\therefore$ $931.6\times.89-1025.8\times.76+720.3-587.3\times.55=50.19$ B.T.U.

Then, $\dfrac{50.19\times778\times43500\times2}{33000\times60}=1715.73$ horse-power in L.P. turbines. Answer.

10. 1st Expansion—
$18+15=33$ lbs. $=255.9+461=716.9^\circ$; 934° = latent heat at 33 lbs.
$11+15=26$ lbs. $=242.3+461=703.3^\circ$; 943.7 = latent heat at 26 lbs.
$\therefore$ $934\times.81-943.7\times.8+716.9-703.3=15.18$ B.T.U. Answer.

2nd Expansion—
$11+15=26$ lbs. $=242.3+461=703.3^\circ$.
$5+15=20$ lbs. $=228+461=689^\circ$; 953.8 latent heat at 12 lbs.
$\therefore$ $943.7\times.8-953.8\times.79+703.3-689=15.76$ B.T.U. Answer.

ɪ3.ɪ + 46ɪ = 67
74.ɪ = ɪ6.24 B

ɪ97.8 + 46ɪ = 6
658.8 = ɪ6.55

ɪ = 64ɪ°.
64ɪ = ɪ7.97 B

46ɪ = 6ɪ8.9.
6ɪ8.9 = 20.27

5ɪ = 602.6.
· 602.6 = ɪ7.75

6ɪ = 587.3.
· 587.3 = ɪ7.56

B.T.U. Total

84 ft. lbs. Aı

822.ɪ; 859.6° =
99; 875.9° = la
99 = 25.94 B.T

3°.
73 = 27.69 B.T

48.ɪ.
8.ɪ = 27.9ɪ B.T

ɩpansion—

$\cdot + 15 = 35$ lbs. $= 259.3 + 461 = 720.3$.

$2 \times .84 - 931.6 \times .82 + 748.1 - 720.3 = 29.96$ B.T.U. Answer.

Then
$$\begin{array}{r} 25.94 \\ 27.69 \\ 27.91 \\ 29.96 \\ \hline 111.50 \end{array}$$
111.50 B.T.U. total expansion. Answer.

pansion—

$0 + 15 = 155$ lbs. $= 361.1 + 461 = 822.1°$.

$0 + 15 = 115$ lbs. $= 338 + 461 = 799$.

$9.6 \times 1 - 875.9 \times .98 + 822.1 - 799 = 24.31$ B.T.U. Answer.

xpansion—

$+ 15 = 80$ lbs. $= 312 + 461 = 773°$.

$5.9 \times .98 - 894.3 \times .96 + 799 - 773 = 25.85$ B.T.U. Answer.

ɩpansion—

$\cdot + 15 = 55$ lbs. $= 287.1 + 461 = 748.1$.

$4.3 \times .96 - 912 \times .945 + 773 - 748.1 = 21.58$ B.T.U. Answer.

ɩpansion—

$\cdot + 15 = .35$ lb. $= 259.3 + 461 = 720.3$.

$2 \times .945 - 931.6 \times .92 + 748.1 - 720.3 = 32.56$ B.T.U. Answer.

Then,
$$\begin{array}{r} 24.31 \\ 25.85 \\ 21.58 \\ 32.56 \\ \hline 104.30 \end{array}$$
Total expansion 104.30 B.T.U. Answer.

pansion—

$+ 15 = 50$ lbs. $= 281° + 461 = 742$.

$+ 15 = 37$ lbs. $= 262.6 + 461 = 723.6$.

$6.3 \times .9 - 929.3 \times .88 + 742 - 723.6 = 25.286$ B.T.U. Answer.

xpansion—

$+ 15 = 37$ lbs. $= 262.6 + 461 = 723.6$.

$+ 15 = 25$ lbs. $= 240.1 + 461 = 701.1$.

$9.3 \times .88 - 945.3 \times .869 + 723.6 - 701.1 = 18.8183$ B.T.U. Answer.

ɩpansion—

$+ 15 = 25$ lbs. $= 240.1 + 461 = 701.1$.

.bs. $+ 15 = 15$ lbs. $= 213.1 + 461 = 674.1$.

$5.3 \times .869 - 964.3 \times .85 + 701.1 - 674.1 = 28.810$ B.T.U. Answer.

:pansion—

$+ 15 = 15$ lbs. $= 213.1 + 461 = 674.1$.

in. vacuum $= 9$ lbs. $= 188.3 + 461 = 649.3$.

$\therefore 964.3 \times .85 - 981.9 \times .839 + 674.1 - 649.3 = 20.64$ B.T

$$\begin{array}{r} 25.286 \\ 18.8183 \\ 28.810 \\ 20.64 \\ \hline 93.55 \end{array}$$ B.T.U. total heat dro

Then, $\dfrac{93.55 \times 778 \times .70 \times 70000}{33000 \times 60} = 1801.1$ horse-pow

14. 1st Expansion—

14 in. vacuum = 8 lbs. = $182.9 + 461 = 643.9°$; latent h
16 in. vacuum = 7 lbs. = $176.9 + 461 = 637.9°$; latent h

$\therefore 985.7 \times .836 - 990 \times .825 + 643.9 - 637.9 = 13.29$ B.T.

2nd Expansion—

16 in. vacuum = 7 lbs. = $176.9 + 461 = 637.9°$.
17 in. vacuum = 6.5 lbs. = $173.6 + 461 = 634.6°$.

$\therefore 990 \times .825 - 992.3 \times .812 + 637.9 - 634.6 = 14.302$ B.T

3rd Expansion—

17 in. vacuum = 6.5 lbs. = $173.6 + 461 = 634.6°$.
19 in. vacuum = 5.5 lbs. = $166.4 + 461 = 627.4°$.

$\therefore 992.3 \times .812 - 997.4 \times .802 + 634.6 - 627.4 = 13.03$ B.

4th Expansion—

19 in. vacuum = 5.5 lbs. = $166.4 + 461 = 627.4°$.
21 in. vacuum = 4.5 lbs. = $157.9 + 461 = 618.9°$.

$\therefore 997.4 \times .802 - 1003.4 \times .792 + 627.4 - 618.9 = 13.72$ B.

5th Expansion—

21 in. vacuum = 4.5 lbs. = $157.9 + 461 = 618.9°$.
23 in. vacuum = 3.5 lbs. = $147.7 + 461 = 608.7°$.

$\therefore 1003.4 \times .792 - 1010.6 \times .78 + 618.9 - 608.7 = 16.624$

6th Expansion—

23 in. vacuum = 3.5 lbs. = $147.7 + 461 = 608.7°$.
24 in. vacuum = 3 lbs. = $141.6 + 461 = 602.6°$.

$\therefore 1010.6 \times .78 - 1015 \times .765 + 608.7 - 602.6 = 17.89$ B.T

7th Expansion—

24 in. vacuum = 3 lbs. = $141.6 + 461 = 602.6°$.
26 in. vacuum = 2 lbs. = $126.3 + 461 = 587.3°$.

$\therefore 1015 \times .765 - 1025.8 \times .755 + 602.6 - 587.3 = 17.29$ B.

8th Expansion—

26 in. vacuum = 2 lbs. = $126.3 + 461 = 587.3°$.
27 in. vacuum = 1.5 lbs. = $115.9 + 461 = 576.9°$.

Then, each L.P. turbine gets $\frac{1340}{2} = 670$ lbs. steam.

$\therefore \frac{277.418 \times 778 \times .58 \times 670}{33000} = \Big\{$ 2541 horse-power in each L.P. turbine. Answer.

Then, $670 \times 60 \div 2541 = 15.8$ lbs. of steam per horse-power per hour. Ans.

18. $145 + 15 = 160$ lbs. $= 363.6 + 461 = 824.6$; latent heat $= 857.8°$.
27 in. vacuum $= 1\frac{1}{2}$ lbs. $= 115.9 + 461 = 576.9$; latent heat $= 1033.2°$.
$\therefore 857.8 \times .99 - 1033.2 \times .76 + 824.6 - 576.9 = 311.690$ B.T.U.
Then, $8000 \times 33000 \div 2000 \times 311.69 \times 778 = .54$ efficiency. Answer.
And $2000 \times 60 \div 8000 = 15$ lbs. steam per horse-power per hour. Ans.

19. $140 + 15 = 155$ lbs. $= 361.1 + 461 = 822.1$.
27 in. vacuum $= 1\frac{1}{2}$ lbs. $= 115.9 + 461 = 576.9$.
$\therefore 859.6 \times .98 - 1033.2 \times .75 + 822.1 - 576.9 = 312.708$ B.T.U

Then, $\frac{5.2 \times 2240}{60} = 194.13$ lbs. of coal per minute,
and as 8.5 lbs. of water are evaporated per lb. of coal,
$\therefore 194.13 \times 8.5 = 1650.1$ lbs. of steam used per minute.
Then, $33000 \times 6500 \div 312.708 \times 778 \times 1650.1 = .53$ efficiency. Answer.

20. $140 + 15 = 155$ lbs. $= 361.1 + 461 = 822.1$.
$21 + 15 = 36$ lbs. $= 260.9 + 461 = 721.9$.
$\therefore 859.6 \times .99 - 930.5 \times .91 + 822.1 - 721.9 = \Big\{$ 104.449 B.T.U. in H.P. turbine.
Again, $20 + 15 = 35$ lbs. $= 259.3 + 461 = 720.3$.
28 in. vacuum $= 1$ lb. $= 102.1 + 461 = 563.1$.
$\therefore 931.6 \times .9 - 1042.9 \times .75 + 720.3 - 563.1 = \Big\{$ 213.465 B.T.U. in L.P. turbines.
Then, total heat drop $= 104.449 + 213.465 = 317.914$ B.T.U. Answer.
$317.914 \times 778 = 247337.09$ ft. lbs. of energy given up. Answer.
Then, $43.2 \times 2240 \div 6 \times 60 = 268.8$ lbs. coal per minute.
$\therefore 268.8 \times 8.5 = 2284.8$ lbs. steam per minute.
And $2284.8 \times 60 \div 10000 = 13.7$ lbs. steam per H.P. per hour. Answer.
$13.7 \div 8.5 = 1.61$ lbs. of coal per horse-power per hour. Answer.
$33000 \times 10000 \div 317.914 \times 778 \times 2284.8 = \Big\{$ .58 efficiency $= 58\%$. Answer.

21. $140 + 15 = 155$ lbs. $= 361 + 461 = 822.1°$; latent heat $= 859.6$.
$20 + 15 = 35$ lbs. $= 259.3 + 461 = 720.3$; latent heat $= 931.6$.
$\therefore 859.6 \times 1 - 931.6 \times .93 + 822.1 - 720.3 = 95.01$.
And $95.01 \times .60 = 57$ B.T.U. in H.P. turbine. Answer.
$20 + 15 = 35$ lbs. $= 259.3 + 461 = 720.3°$; latent heat $= 931.6°$.
$1\frac{1}{2}$ lbs. $= 115.9 + 461 = 576.9°$; latent heat $= 1033.2°$.
$\therefore 931.6 \times .93 - 1033.2 \times .78 + 720.3 - 576.9 = 203.892$.
$\therefore \frac{203.892 \times .60}{2} = 61.16$ B.T.U. in each L.P. turbine. Answer.

d expansion blades $= .875 \times 1.4 = 1.22$ in., say $1\frac{1}{4}$ in.
d ,, $= 1.25 \times 1.4 = 1.75$,, $1\frac{3}{4}$,,
, ,, $= 1.75 \times 1.4 = 2.45$,, $2\frac{1}{2}$,, } Ans.

Drum ratio $= 60$ in. $\div 42$ in. $= 1.42$.

st expansion blades $= \dfrac{2.5 \times 1.4}{1.42 \times 1.42 \times 2} = .86$ in., say $\frac{7}{8}$ in.
nd ,, $= .875 \times 1.4 = 1.22$ in., say $1\frac{1}{4}$ in.
rd ,, $= 1.25 \times 1.4 =$ $1\frac{3}{4}$,,
th ,, $= 1.75 \times 1.4 = 2.45$,, $2\frac{1}{2}$,,
th ,, $= 2.5 \times 1.4 =$ $3\frac{1}{2}$,,
th ,, $= 3.5 \times 1.4 = 4.9$,, 5 ,,
th ,, = 5 ,,
th ,, = 5 ,, } Ans.

am flow at 7th expansion $= \dfrac{5000 \times 14 \times 90}{2 \times 60 \times 60} = 875$ cub. ft.

lear area between blades $= 875 \div 350 = 2.5$ sq. ft.

$$\text{Annulus area} = \frac{(60 + 5) \times 3.1416 \times 5}{144} = 7 \text{ sq. ft.}$$

Then, annulus factor $= 7 \div 2.5 = 2.8$.

am flow at 8th expansion $= \dfrac{5000 \times 14 \times 140}{2 \times 60 \times 60} = 1361$ cub. ft.

Required clear area $= 1361 \div 380 = 3.58$ sq. ft.
Annulus area $= 7$ sq. ft. as formerly.
Then, annulus factor $= 7 \div 3.58 = 1.95$. Ans.

$$\frac{8000 \times 13.5 \times 2.78}{60 \times 60} = 83.4 \text{ cub. ft. flow per sec.}$$

$$\frac{2700}{\sqrt{156}} = 217 \text{ ft. per sec. steam velocity at 1st expansion.}$$

$$\frac{83.4}{217} = .384 \text{ ft.} = \text{required clear exit area between blades.}$$

$$\sqrt{\frac{.384 \times 3 + 8.295}{.7854}} = 3.467.$$

$$\frac{3.467 - 3.25}{2} = .108 \text{ ft. blade height at 1st expansion.}$$

$.108 \times 12 = 1.29$ in., say $1\frac{1}{4}$ in. height of blades 1st expansion.
$1.25 \times 1.4 = 1.75$,, $1\frac{3}{4}$,, 2nd ,,
$1.75 \times 1.4 = 2.45$,, $2\frac{1}{2}$,, 3rd ,,
$2.5 \times 1.4 = 3.5$,, $3\frac{1}{2}$,, 4th ,,

Drum ratio $= 55 \div 39 = 1.41$.

hts $= \dfrac{3.5 \times 1.4}{1.41 \times 1.41 \times 2} = 1.23$ in., say $1\frac{1}{4}$ in. at 1st L.P. expansion.

Shaft Horse-Power.

$$\frac{1.53 \times .68 \times 6.75^4 \times 600}{12.5 \times 36} = 2879 \text{ horse-power.}$$

$$\frac{1.53 \times .68 \times 6.75^4 \times 650}{12.5 \times 36} = 3119 \text{ horse-power.}$$

$$\frac{.53 \times .68 \times 6.75^4 \times 654}{12.5 \times 36} = 3138 \text{ horse-power.}$$

er $= 2879 + 3119 + 3138 = 9136.$

$9136 \div .9 = 10151$ horse power. Answer.

$$\frac{1.53 \times .8 \times 4.75^4 \times 690}{12.5 \times 33} = 1042 \text{ horse-power.}$$

$$\frac{1.53 \times .8 \times 4.75^4 \times 830}{12.5 \times 33} = 1253 \text{ horse-power.}$$

$$\frac{1.53 \times .8 \times 4.75 \times 825}{12.5 \times 33} = 1246 \text{ horse-power.}$$

$2 + 1253 + 1246 = 3541$ horse-power.

$3541 \div .9 = 3934.4$ horse-power. Answer.

er $\frac{1.53 \times .35 \times 6^4 \times 490}{12.5 \times 25} = 1088$ horse-power.

r $\frac{1.53 \times .35 \times 6^4 \times 580}{12.5 \times 25} = 1288$ horse-power

r $\frac{1.53 \times .35 \times 6^4 \times 582}{12.5 \times 25} = 1292$ horse-power.

$8 + 1288 + 1292 = 3668$ horse-power.

$3668 \div .9 = 4075.5$ horse-power. Answer.

r $\frac{1.53 \times .6 \times 10^4 \times 300}{12.5 \times 60} = 3672$ horse-power.

r $\frac{1.53 \times .6 \times 10^4 \times 300}{12.5 \times 60} = 3672$ horse-power.

r $\frac{1.53 \times .6 \times 10^4 \times 300}{12.5 \times 60} = 3672$ horse-power.

$= 3672 + 3672 + 3672 = 11016.$

$11016 \div .9 = 12240$ horse-power. Answer.

$$\frac{1.53 \times .358 \times 8^4 \times 625}{12.5 \times 40} = 2840 \text{ horse-power.}$$

$$\frac{1.53 \times .358 \times 8^4 \times 630}{12.5 \times 40} = 2826 \text{ horse-power.}$$

$$\frac{1.53 \times .358 \times 8^4 \times 632}{12.5 \times 40} = 2835 \text{ horse-power.}$$

$0 + 2826 + 2835 = 8465$ horse power.

$8465 \div .9 = 9405.5$ horse power. Answer.

The Marine Steam Turbine.

w $\Big\} = \dfrac{(487^2 - 350^2) \times .82 \times 11.25 \times 60}{64.4 \times 33000} = 29.86$ horse-power.

n $\Big\} = 29.86 \times 8 = 238.88$ horse-power.

r $\Big\} = 238.88 \times 33000 \div 778 = 10132.44$ B.T.U.

s $= \left(\dfrac{4.4375 \times 3.1416 \times 650}{60}\right) \div 487 = .310$ ratio.

Ans.

w $\Big\} = \dfrac{(541^2 - 397^2) \times .76 \times 11.25 \times 60}{64.4 \times 33000} = 32.6$ horse-power.

n $\Big\} = 32.6 \times 8 = 260.8$ horse-power.

r $\Big\} = 260.8 \times 33000 \div 778 = 11062.2$ B.T.U.

s $= \left(\dfrac{4.625 \times 3.1416 \times 650}{60}\right) \div 541 = .29$ ratio.

Ans.

w $\Big\} = \dfrac{(600^2 - 475^2) \times .74 \times 11.5 \times 60}{64.4 \times 33000} = 31.58$ horse-power.

n $\Big\} = 31.58 \times 8 = 252.64$ horse-power.

r $\Big\} = 252.64 \times 33000 \div 778 = 10716.09$ B.T.U.

s $= \left(\dfrac{4.66 \times 3.1416 \times 650}{60}\right) \div 600 = .264$ ratio.

Ans.

w $\Big\} = \dfrac{(750^2 - 646^2) \times .715 \times 11.25 \times 60}{64.4 \times 33000} = 32.97$ horse-power.

n $\Big\} = 32.97 \times 8 = 263.76$ horse-power.

r $\Big\} = 263.76 \times 33000 \div 778 = 11187.7$ B.T.U.

s $= \left(\dfrac{4.75 \times 3.1416 \times 650}{60}\right) \div 750 = .215$ ratio.

Ans.

Steam Condensed in Turbines.

$\times (1 - .85) = 112000 \times .15 = 16800$ lbs. Answer.

$6000 \times 13.5 = 81000$ lbs. steam.

$\times (1 - .848) = 81000 \times .152 = 12312$ lbs. per hour. Answer.

$45000 \times \frac{13}{2} = 292500$ lb

$00 \times (1 - .835) = 292500 \times .16$

Then $48262.5 \div 2240 = 21.5$

APPENDIX.

GENERAL NOTES.

Efficiency of "Lusitania" Turbines.—Referring to the paper read by Thomas Bell, Esq. (page 153), on the trial and sea performances of the "Lusitania," the steam consumption of the turbines at an average shaft horse-power of 65,000 is given as 13.1 lbs. per hour. Assuming an H.P. initial gauge pressure of 150 lbs., and a terminal L.P. pressure of 1.5 lbs. absolute, we can determine the over-all turbine efficiency as follows:—

Take adiabatic expansion, with H.P. initial steam of .99 dryness, and L.P. terminal steam of .77 dryness.

Then, $150 + 15 = 165$ lbs. absolute, temperature 366 deg.,
and, $366 + 461 = 827$ absolute temperature,
165 lbs. absolute = 856.2 B.T.U. latent heat,
1.5 " = 115.9 temperature,
and $115.9 + 461 = 576.9$ absolute temperature,
1.5 lbs. absolute = 1033.2 B.T.U. latent heat.

$$\begin{aligned}\text{Heat drop} &= 856.2 \times .99 - 1033.2 \times .77 + 827 - 576.9 \\ &= 847.638 - 795.564 + 827 - 576.9 \\ &= 1372.46 - 1674.63 \\ &= 302.17 \text{ B.T.U.}\end{aligned}$$

$$\text{Efficiency} = \frac{33000 \times 60}{\text{lbs. steam} \times \text{B.T.U.} \times 778} = \frac{33000 \times 60}{13.1 \times 302.17 \times 778} = .64.$$

The turbine efficiency is therefore equal to 64 per cent.

Again, it may be pointed out that as the steam consumption is given per *shaft* horse-power per hour the consumption will be still less per I.H.P., and taking the ratio of B.H.P. to I.H.P. as .93 is to 1 for this special case of large power, then, $13.1 \times .93 = 12.18$ lbs. steam per I.H.P. per hour for turbines alone. The total steam consumption (including all auxiliary gear) is given as 15.35 lbs. per shaft horse-power per hour, and again reducing to the I.H.P. standard we get $15.35 \times .93 = 14.27$ lbs. steam per I.H.P. per hour, which may be considered as a very satisfactory result indeed, and one comparing most favourably with the best modern marine reciprocating engine practice.

experiments with the reduced scale hull models obtain progressive "tow rope" resistances which are the actual resistances at various speeds of the model hull (the propeller being omitted). These are made up as follows:—

Resistance.—1. Skin frictional resistance of the hull surface. 2. Wave making resistance of the hull body. 3. Eddy making resistance of the hull body.

The foregoing constitute what may perhaps be termed the true resistances, and to overcome these the effective horse-power is required.

Power Losses.—The losses of engine power are made up as follows:—

1. Friction (initial and load).
2. Propeller inefficiency.
3. Hull inefficiency.

The frictional losses are those occasioned by the working parts and the power absorbed by the thrust block. The propeller losses are due to excessive slip, blades friction, and other causes, and the hull efficiency is a result which may either be under or above unity, according to the difference between what is called "augmentation of resistance," due to the propeller blades at the stern, and "wake speed gain." Generally, however, the "wake speed gain" balances the augment of resistance to within a very few per cent., although an allowance of about 95 per cent. is often taken as the "hull efficiency."

The "wake speed" is produced by the water closing in on the stern as the hull advances, and this body of water acquires a forward motion or speed varying in degree with the lines of the hull body.

Utilisation of Power.—The total I.H.P. developed by the engines is therefore used up somewhat as follows, although it must be understood that the values given vary in different cases and under different conditions in the same case:—

Taking the total I.H.P. as 100 per cent.

Reciprocating Engines.

Indicated horse-power	100 per cent.
Engine friction loss	10 „
Horse-power at propeller	90 „
Propeller efficiency, 62 per cent.	
Then, $90 \times .62 = 55.8$ „	
Horse-power by propeller	55.8 „
Hull efficiency, 95 per cent.	
Then, $55.8 \times .95 = 53$ „	
Effective horse-power	53 „

Therefore propulsive efficiency $= \frac{53}{\ }$, or .53.

rse turbines are incorporated. This might be over-
uction pipe to the dry air pump from the reverse
ising turbines, to be open when running ahead full
nt would, however, have the disadvantage of com-
s, and would also act as a hindrance to quick re-
Turbine over-all efficiency would undoubtedly be
a of, say, from 28 in. or 29 in. could be obtained in

bines when opened up after Service.—If the
working order and priming prevented, the turbines,
nning, do not as a rule show any evidence of corrosion
noted recently by the writer, the turbines were
dition, no marks of any kind being visible on the
, or blading. To obtain this desirable result, the
attended to, and all tendency to priming checked
n the case referred to above did not exceed a
r gallon.

; are often in use (as in the case of channel steamers),
ust block rings are found to wear much more than
; that the propeller thrust when running astern is
thrust on the reverse turbine blades. The astern
y screwing the top half of the thrust cover forward
; stud and gear. (See page 107.)

If the steam expansion is only partly saturation and partly adiabatic (which condition often exists in actual practice), then the curve does not extend so far to the right as the saturation curve, but is nearer to the vertical line of adiabatic expansion.

Base Line.

It is usual to take as a base line a zero temperature of 32° Fahr. and show from this temperature upwards the entropy changes which occur with rise or fall of temperature, etc. Again it must be noted that all temperatures are absolute, that is—

Temperature Farenheit + 461° = Absolute Temperature.

Scale of Entropy.

A scale of entropy must be fixed upon, such as ·1 entropy = ½ in. or ¼ in. of length, horizontally.

Increases of, say, 50° Fahr. or 100° Fahr., may be taken by scale, such as 50° per inch, and each increase worked out for the plotting out of the curves.

During the heating up of water from, say, the feed temperature to the steam temperature, the diagram developes a flat curve with increase of entropy, but when the boiling point is reached, as no further increase of temperature takes place while heat is still being supplied, the diagram then shows a horizontal line with increase of entropy.

During expansion the diagram developes a vertical line if the expansion is adiabatic in nature, and a curve to the right if the expansion is that of saturated steam.

Cylinder condensation affects the diagram by pulling over the expansion line to the left of the adiabatic, as heat is then being lost, and the useful work done reduced.

During condensation, or during constant temperature compression, the line of the diagram is horizontal, as heat is then being removed and the entropy decreased.

During constant volume condensation, the line is curved, and falls away to the left.

Referring to diagram No. 1, the area A B C D is that available for work theoretically, but in practice various losses occur which diminish this to about one-half.

The curve C K is that obtained when heat is supplied to keep the steam up to saturation conditions.

The vertical line C D is that produced by adiabatic expansion (no heat given to, or taken from).

From A to B shows rise of temperature and increase of entropy.

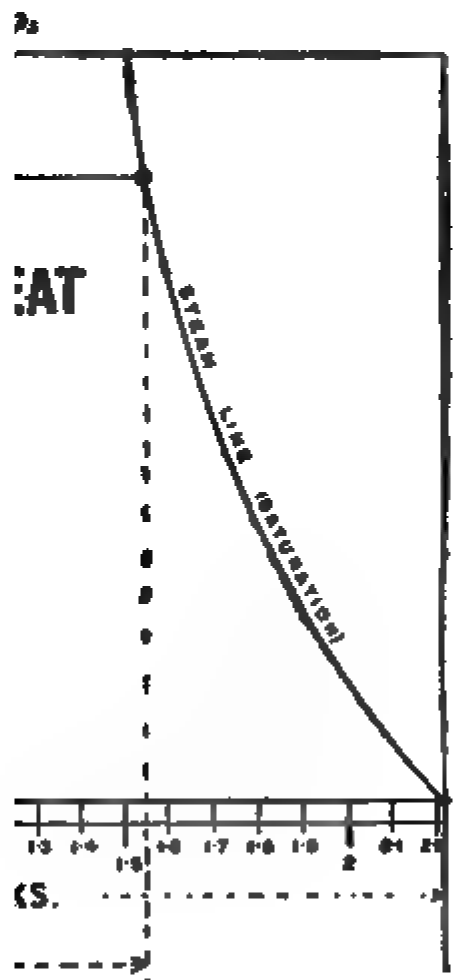
From B to C shows constant (isothermal) temperature, producing evaporation by latent heat, also increase of entropy.

'urbine.

due to adiabatic (constant

am or steam compression
ing taken out. the entropy

ater at 32°; line 2 shows
ture: line 3 shows entropy



4 shows entropy value for
uring adiabatic expansion;
on expansion.
se again and complete the

becomes less dry, and the
K (for adiabatic expansion)

$= \frac{AD}{AK}$.

The heat unit area of the "heat rejected" portion is found by multiplying height in degrees by mean width in entropy.

When once the diagram is completed as described, the lines forming it are constant for all problems, so that the same chart holds good and can be used repeatedly, the various problems being worked out by the help of tracing paper laid over the master diagram, and the lines of which can then be traced through as required. It is, however, good practice for the student to plot out one or two charts, using a large scale for the purpose, such as, say, 1 in. for 50° temperature, and 1 in. for 2 units entropy.

Areas of Diagram.

As height of diagram represents temperature, and horizontal width represents entropy,

Then, $$\text{Entropy} = \frac{\text{Heat Units}}{\text{Temperature}}.$$

Then, $$\underset{\text{(Temp. diff.)}}{\text{Height}} \times \underset{\text{(Entropy)}}{\text{Width}} = \text{Heat Units.}$$

Again, $$\text{Heat Units} \times 778 = \text{Foot-lbs. of Work.}$$

Example 2. Find the heat unit value of the area contained between the temperatures of 40° Fahr. and 380° Fahr., the mean entropy being 1·868.

Then, $$\text{Heat Units} = (380° - 40°) \times 1{\cdot}868 = 635.$$

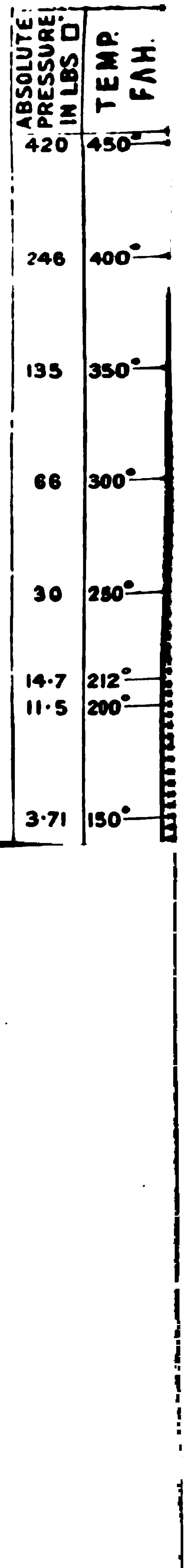
$$\text{Foot-lbs.} = 635 \times 778 = 494030.$$

When the area is not a rectangle, such as that found in the upper part of the diagram, the mean width *measured in entropy* must be found by any convenient method, such as those previously mentioned, and this multiplied by the height *measured in degrees* will give the necessary area measurement in heat units.

In practice condensation is usually effected more or less at constant volume with fall of temperature, and this shows on the entropy diagram as a curve falling away from the release temperature of 200° to the left. This represents an increased percentage of heat rejected as shown above, and the thermal, or heat, efficiency is therefore proportionally reduced.

Saturation Curve.

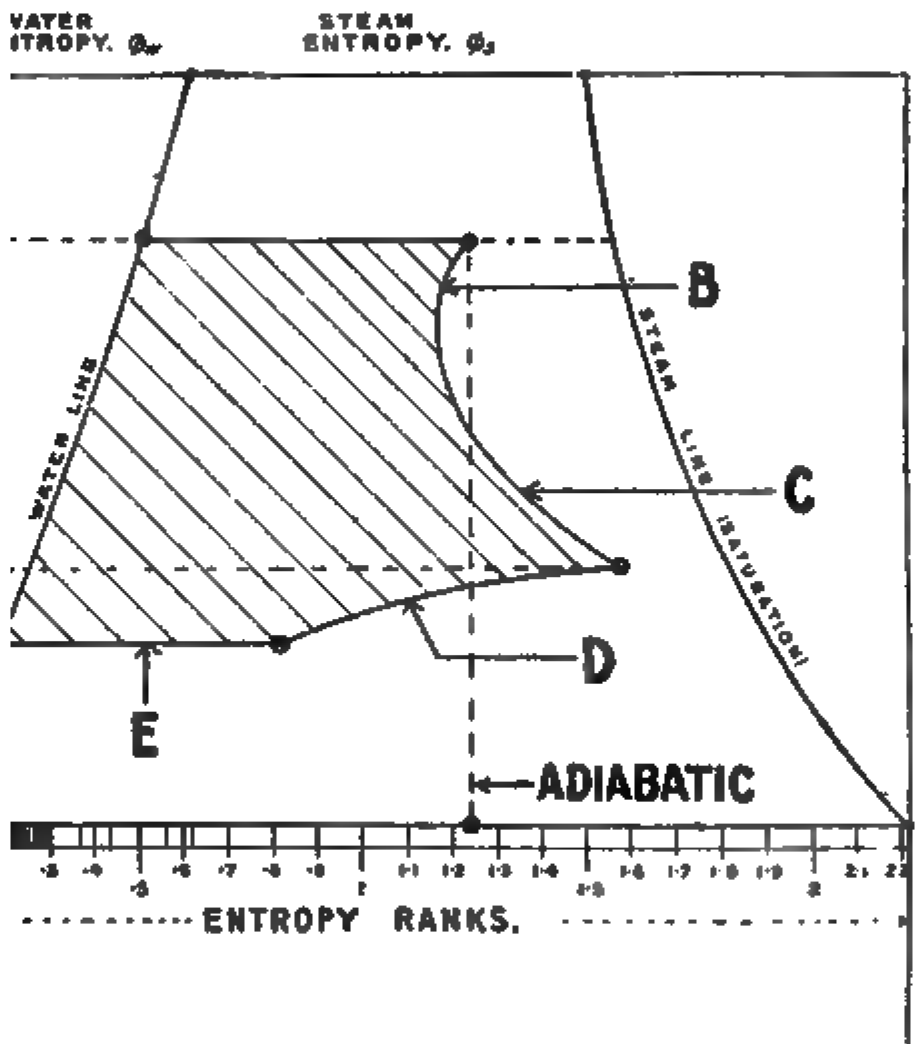
If the pressure and temperature of the steam during expansion agree with the figures given in the t
in the condition known as dry, or, as i
the curve follows out the line as sho
is explained on p. 334. In actual



cylinders well lagged show of approximate adiabatic expansion taking place.

If, however, part of the steam is condensed on the cylinder walls over and above that portion condensed by work done, then the
l fall over to the left, as shown at D, and the
reased in proportion.

tice condensation occurs in the cylinder at the



No. 6.

;team Turbine.

er at 160° is heated to 380°, then
panded adiabatically to 140°; find
ıgram No. 11).

and steam curves as in previous
Iethod 2, as follows:—

601° absolute.

;7° mean temperature.

$\frac{- 32}{;7} = \cdot 197.$

: 621° absolute.

mean temperature.

$\frac{140°}{\text{I}} = \cdot 0327.$

erature 250°.

711° absolute.

5° mean temperature.

and $\cdot 136 \times 1{\cdot}017 = \cdot 138.$

absolute temperature.

mean temperature.

and $\cdot 167 \times 1{\cdot}04 = \cdot 173.$

found for each heat, increase to
point as shown, and draw in by
ɔoints so marked.

alue of one pound steam at 380°
s (p. 357), by the steam tempera-
e of which set off to the *right* of

bles) = 845 B.T.U.

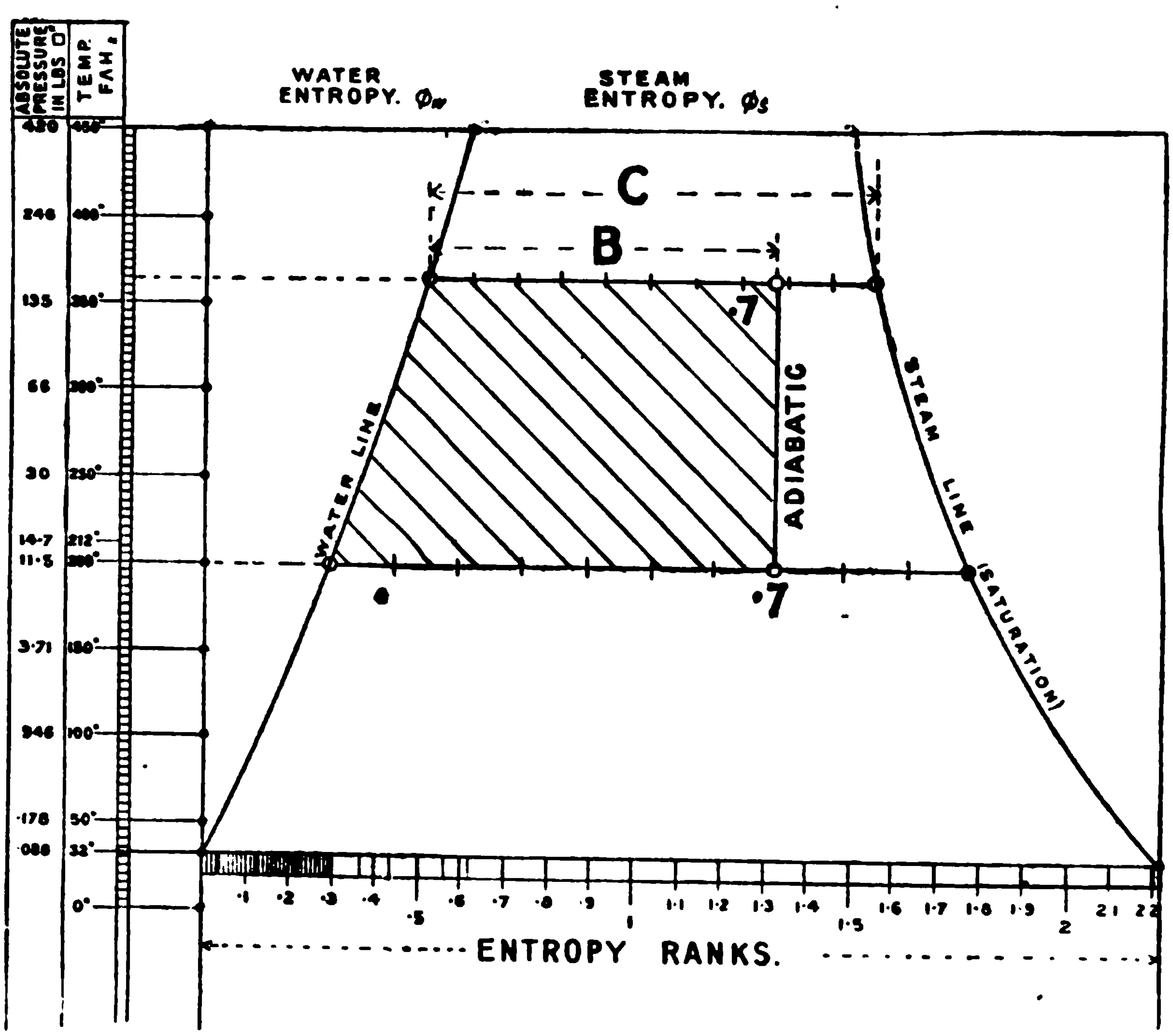
$= \frac{845}{841} = 1{\cdot}005.$

s 250°, 160°, 140°, as here shown.

: value = 938 B.T.U.
,, = 1002 ,,
,, = 1016 ,,

Example 7.—A pound of feed water at 200° temperature is evaporated into steam of 165 lbs. gauge pressure, and with a dryness of ·9; during expansion in the cylinder, and at 350° temperature the dryness is ·70, and at 300° the dryness is ·95; from this point down to exhaust at 160° the expansion is adiabatic in nature. Show the above effects on an entropy diagram. (Diagram No. 12.)

Draw horizontal lines across the diagram at the temperature levels required: these are 372°, 350°, 300°, 200°, and 160°.



No. 13.

At 372° level mark off ·9 of the length *ab* to represent the initial dryness of the steam.

At 350° level mark off ·7 of *cd* to represent the dryness at that stage of expansion.

At 300° level mark off ·95 of *ef* to represent the dryness at that stage of expansion. Draw in by hand a curve connecting the three points, and drop a vertical line from the last one down to the 160° level to complete the diagram.

Example 8.—A pound of feed water 150° temperature is heated and evaporated into wet steam of 150 lbs. gauge pressure; after adiabatic expansion to a pressure of 11·5 lbs. absolute, the dryness

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py

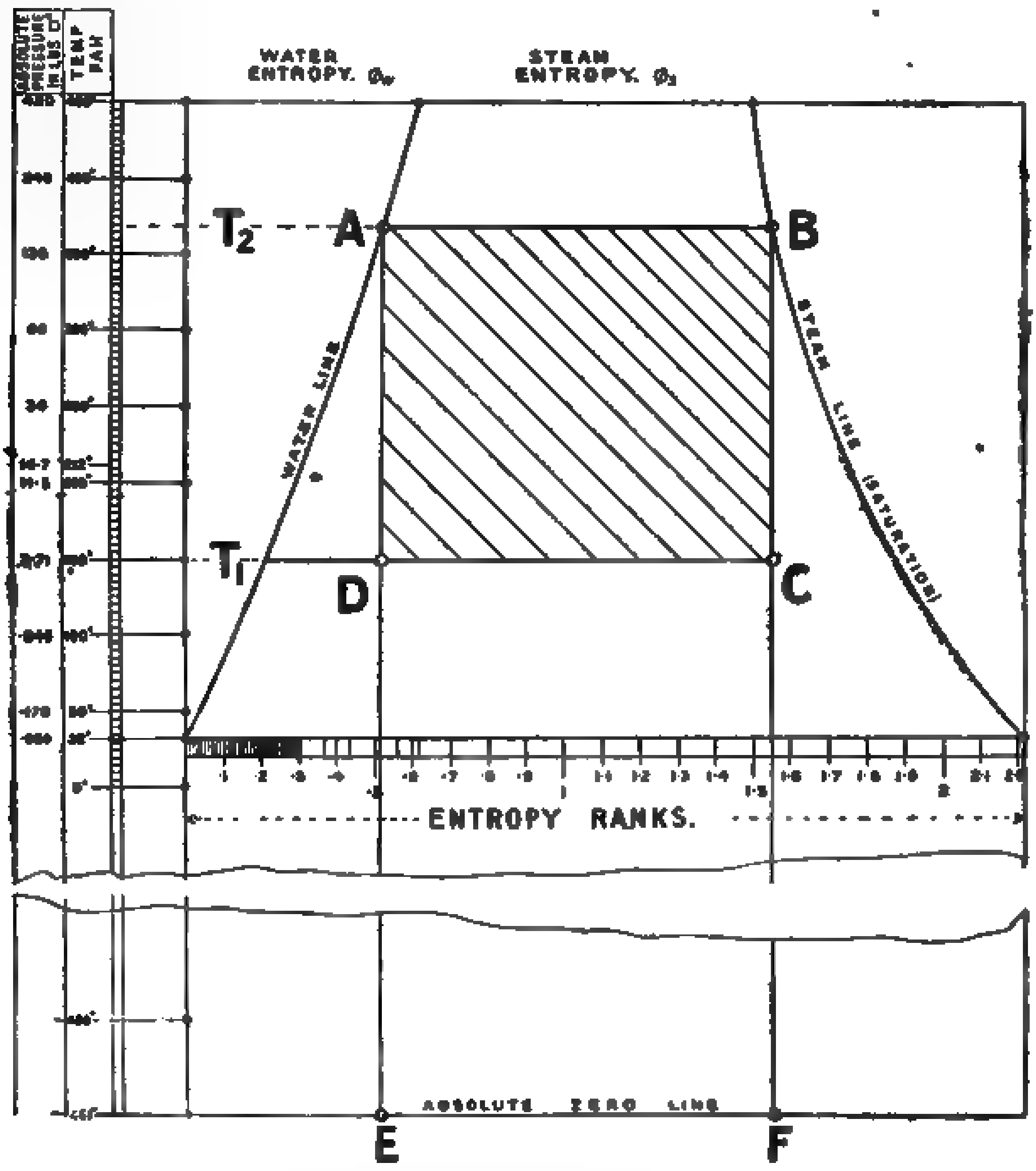
of

ra-

2. Adiabatic expansion of the steam down to temperature T_2 from B to C.

3. Isothermal compression at T_2 from C to D.

4. Adiabatic compression from D to A, that is, from T_1 back to T_2.



No. 16.

Work Done area = ABCD.
Heat Supplied area = EABF.
Efficiency Ratio $= \dfrac{ABCD}{EABF}$.
Heat Rejected area = EDCF.
Heat Units available (per pound) = AB (or DC) $\times (T_1 - T_2)$.

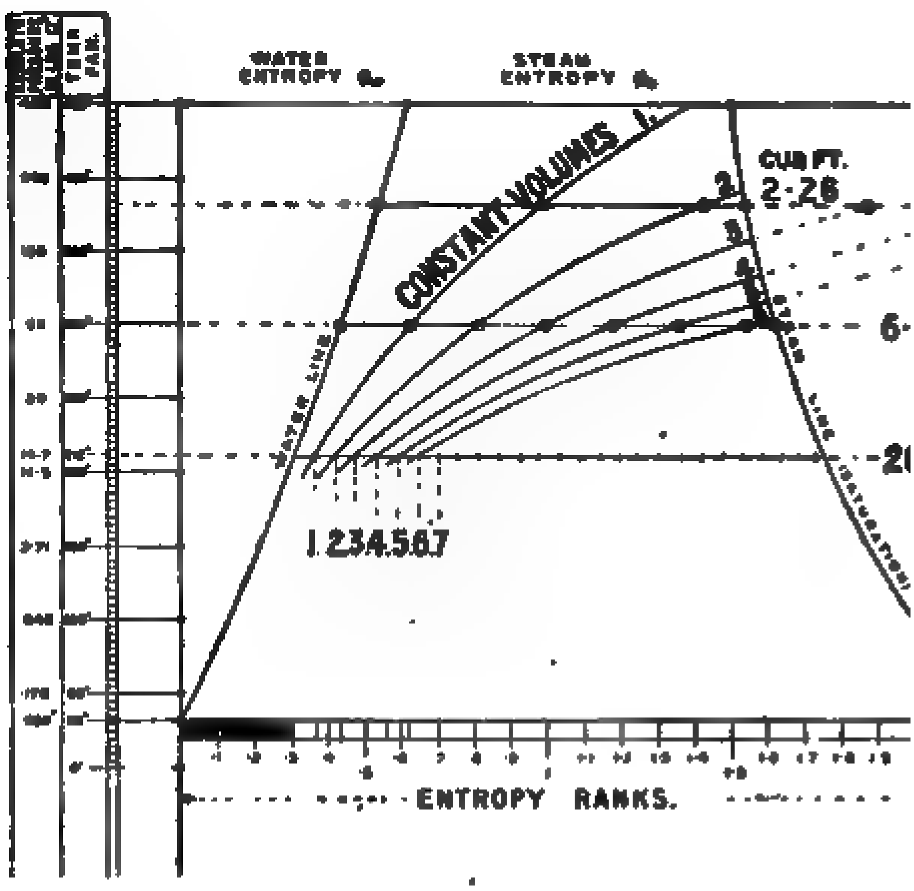
Therefore, **Efficiency** $= \dfrac{T_2 - T_1}{T_2}$.

Constant Volume Lines. (Diagrams No. 8 and No

By constant volume lines are meant the lines entropy diagram when the steam condenses at fall and entropy, but with the same *volume.* This condensers of engines, in which the space occupie remains the same all the time, owing to the regu which goes on taking out the heat of the incoming st

To show then on the $\theta\phi$ diagram, actual conder volume lines are necessary for even approximate accu results.

The volumes lines may be constructed as follows :



No. 17.

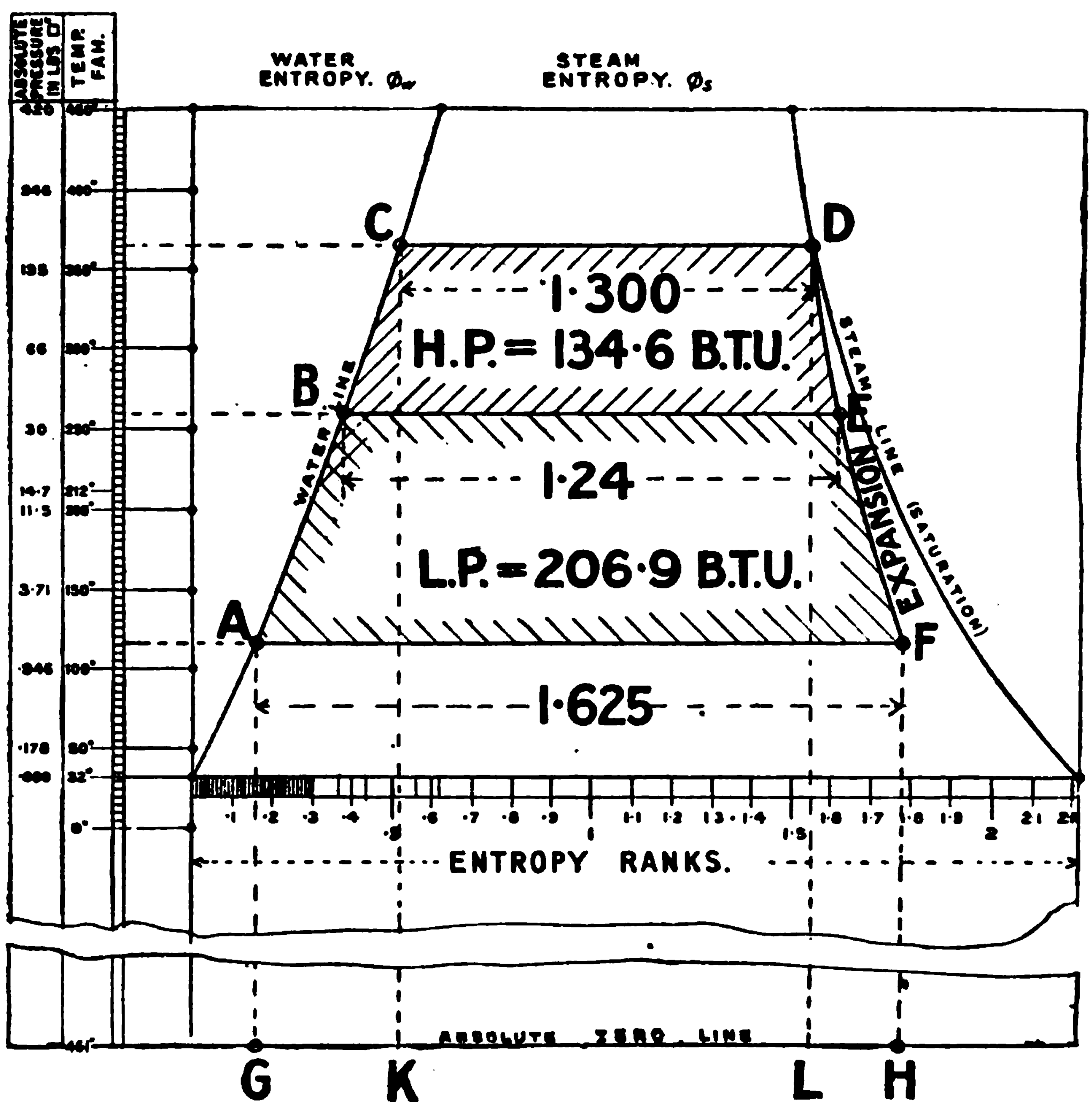
1. Select a few suitable pressures, say three or down opposite, the corresponding specific volumes a taken from the steam table as shown below.

Pressure.	Temperature Fahr.	Specific Vol
200 lbs. absolute	382°	2·26 cub.
67 „ „	300°	6·31 „
14·7 „ „	212°	26·36 „

2. Referring to the diagram No. 17, divide up by trial th horizontal temperature line of 382° into 2·26 divisions representin cubic feet.

3. Divide up the 300° temperature line into 6·3 divisio representing cubic feet.

4. Divide up the 212° temperature line into 26·3 divisio representing cubic feet.



No. 18.

5. Connect up by hand, and afterwards by means of wood "sweeps," the corresponding volumes for each pressure temperatu as shown; that is, join up 1 of 382° with 1 of 300° and 1 of 212°, 2 of 382° with 2 of 300°, and 2 of 212°, and so on: when th joined up the lines are those of constant volume. The consta volume lines show the drop in entropy produced for any give pressure and condensation temperature.

L.P. Turbines.

$206 \cdot 9 \div 2 = 103 \cdot 45$ B.T.U. drop per each L.P. turbine.

And, $103 \cdot 45 \div 6 = 17 \cdot 24$ B.T.U. drop per expansion.

Also, $17 \cdot 24 \div 10 = 1 \cdot 724$ B.T.U. per pair of blade rows.

Exercises.

1. Find ranks entropy required to heat 1 lb. water at 60° temperature Fahr. up to 200° temperature Fahr.
   Answer. ·239 ranks ϕ.

2. 1 lb. water at 212° temperature is evaporated into steam of the same temperature. Find the entropy required.
   Answer. 1·436 ranks ϕ.

3. 1 lb. steam at 212° temperature is expanded under saturation conditions down to 102° temperature. Find the entropy required to keep the steam on the saturation curve line during expansion. Answer. ·416 ranks ϕ.

4. If the Fahr. temperature of 1 lb. saturated steam is 371° and the entropy ranks 1·025, find the latent heat value in B.T.U.
   Answer. 852·8 B.T.U.

5. 1 lb. water at 32° temperature is heated up to 40° temperature. Find the entropy. Answer. ·016 ranks.

6. 1 lb. water at 120° is heated up to 140°. Find the increase of entropy. Answer. ·0339 ranks ϕ.

7. 1 lb. water at 32° Fahr. is heated up to 360°, and then evaporated into steam. Find the water entropy (ϕ_w) and the steam entropy (ϕ_s) if the expansion down to 200° Fahr. is adiabatic.
   Answer. $\begin{cases} \cdot 5169\ \phi \\ 1 \cdot 048\ \ \phi_s \end{cases}$

8. Referring to question 7, find the increase of entropy if the steam remained saturated during expansion. Also express the total entropy of water and steam under these conditions.
   Answer. $\begin{cases} \cdot 424\ \phi_s \\ 1 \cdot 765\ \ \phi. \end{cases}$

9. 1 lb. water at 32° Fahr. is heated up to 137° Fahr. Find the entropy. Answer. ·192 ranks ϕ.

10. 1 lb. saturated steam at 380° Fahr. is superheated at constant pressure up to 500° Fahr. Find the increase of entropy required to produce the superheat. Answer. ·0638 ranks ϕ.

11. The vertical scale of an entropy diagram is equal to 1 in. to 45° Fahr., and the horizontal scale, 1 in. to ·2 entropy ranks. Find the value in B.T.U. and in foot-lbs. of 1 sq. in.
    Answer. 9 B.T.U. 7002 foot-lbs.

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T.U. valu

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is afterwa
ally down

f 1 lb. sat
50° Fahr.,
c. Also fi
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ram for th
(1) the entropy at the feed tempe
and (3) after evaporation, the expan
determine (4) the thermal efficienc
units utilised as work, and (6) th
condenser per lb. of water evaporate

Data. Feed temperature 141°, steam
pressure 175 lbs.), condensation at constan

Answers. {Entropy, ·198 ; en
Efficiency, ·25 ; v
rejected units, 8

16. Find (1) the entropy value of the wat
at steam temperature, and (3) the
steam after evaporation and after
express the thermal efficiency and g
of work available from this efficienc

Data. Feed temperature 200°, steam
temperature 388°, expansion on saturation
constant temperature of 200°.

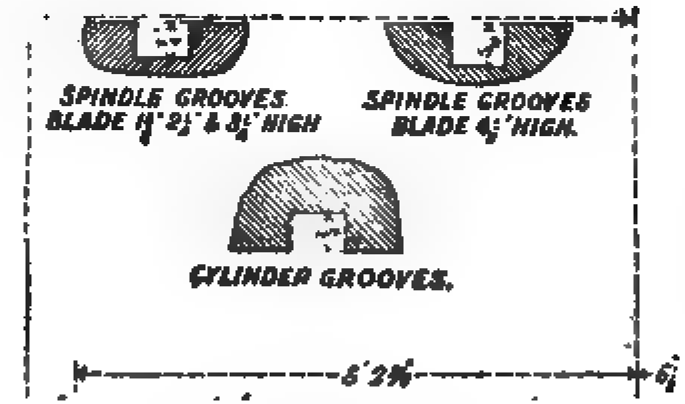
Answers. {Entropy, ·294
1·538 ; entro
Efficiency, ·19 ;

17. In a marine turbine the H.P. initial
the condition of the steam being
pressure is 26 in. vacuum (say .
expansion is adiabatic. Find th
available, and the final dryness of th

Entropy of Water, from 32° Fahr. to 400° Fahr.

Note.—ϕ_w = Entropy of water.

Temperature ° Fahr. t.	Absolute Temperature. T. $t. + 461$	Specific Heat. s.	Total Entropy above Water at 32° Fahr. ϕ_w.
32	493	...	...
40	501	1·0	0·016
50	511	1·0	[illegible]
60	521	1·0	0·055
70	531	1·001	0·074
[illegible]	[illegible]	1·001	0·093
[illegible]	[illegible]	[illegible]	0·111
100	561	1·002	0·129
110	571	1·003	0·147
120	581	1·004	0·164
[illegible]	591	1·004	0·181
140	601	1·005	0·198
150	611	[illegible]	0·215
160	621	1·007	0·231
170	[illegible]	1·008	0·247
[illegible]	641	[illegible]	[illegible]
190	651	1·010	0·279
200	[illegible]	1·011	0·294
210	671	1·012	0·310
220	[illegible]	1·013	0·325
230	691	1·014	0·339
240	701	1·015	0·354
250	711	1·017	[illegible]
260	[illegible]	1·019	0·383
[illegible]	731	[illegible]	[illegible]
[illegible]	741	[illegible]	0·411
290	751	1·024	0·424
[illegible]	[illegible]	[illegible]	0·438
310	771	1·027	0·451
320	781	1·029	0·465
330	791	1·031	0·478
340	801	1·032	[illegible]
350	[illegible]	1·034	0·504
360	[illegible]	[illegible]	0·516
370	831	[illegible]	0·529
380	841	1·040	[illegible]
390	[illegible]	[illegible]	[illegible]
[illegible]	861	1·044	0·566

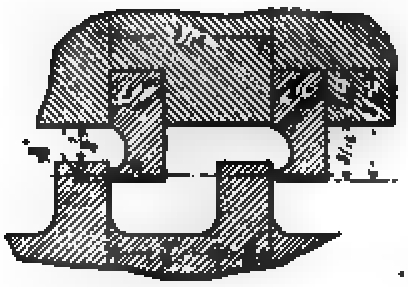


No. 1.—Parson's Type H.P. Reaction Turbine Complete.*

The increase of blade height is approximately in the ratio of as $1 : \sqrt{2}$ or as $1 : 1{\cdot}41$.

DATA.

S.H.P. - - -	20000 (trial).	1 H.P. turbine (centre).
Speed - - -	18·5 knots (trial).	2 L.P. turbines (wing).
Revolutions per minute -	290 (trial).	2 astern turbines (in L.P. casings).
3 shafts, each with 1 propeller.		



DETAIL OF DUMMY STRIP.

No. 6.—L.P. Ahead Dummy.*

Consisting of 20 rings of brass strip, $\frac{3}{16}$ in. thick, and $\frac{9}{16}$ in. pitch. The steam is leaking through the clearance of, say, $\frac{25}{1000}$ in. or ·025 in. alternately wire draws and expands, thus falling continuously in pressure, ring after ring, the final pressure dropping to below that of the atmosphere.

Note.—When superheated steam is used the undercut strips are formed of copper instead of brass.

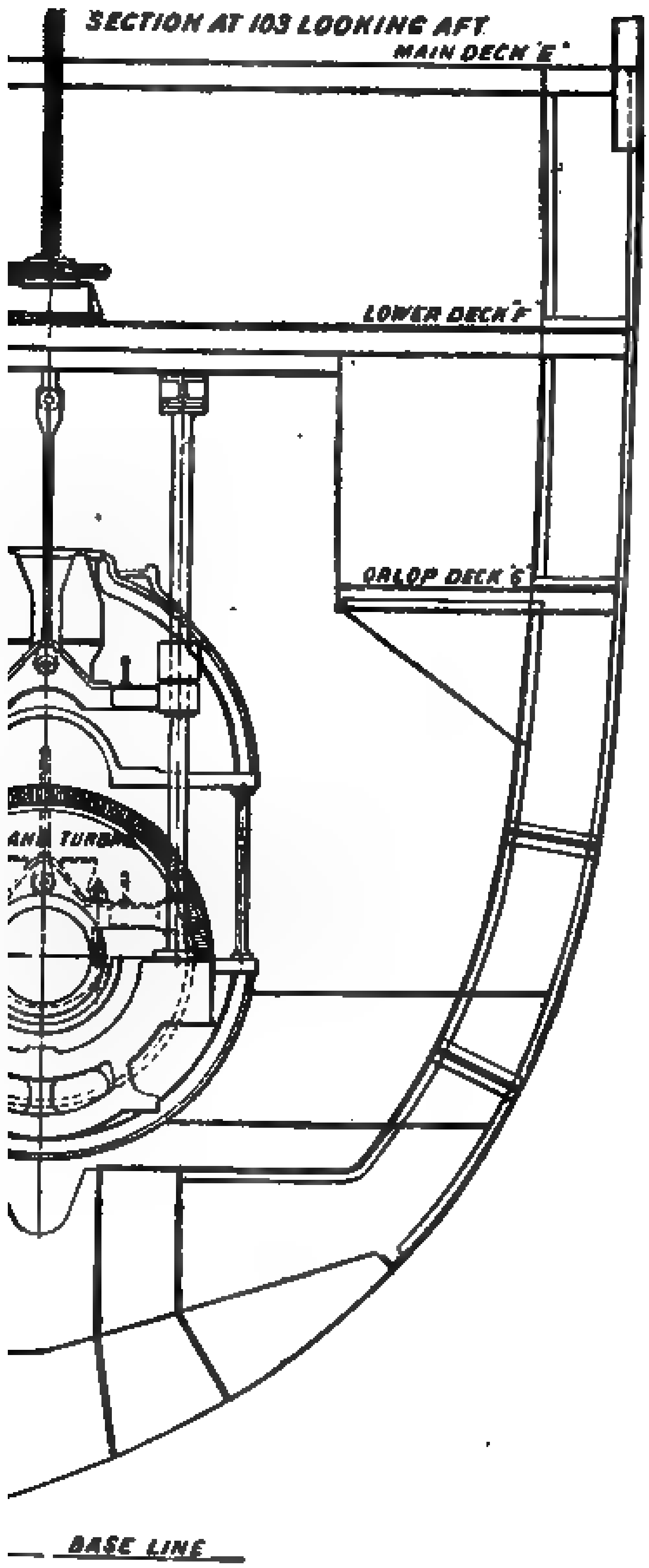
No. 7.—Rotor Spindle Glands.*

Consisting of 13 double "fin" rings, combined with 5 ramsbottom rings, the gland case being fitted with "leak off" and "steam to gland" pockets.

No. 8.—Astern Turbine Dummy.*

Consists of 8 double rows of "fin" rings, through which the leaking steam is wire drawn and reduced in pressure.

Marine Steam Turbine.



.—Turbine Lifting Gear.*
T.S.S. "Aqui

Reprinted from E

h of **Blades** (11) means the distance between the blade row of any one expansion measured on the rotor longitudinally.

Clearance (11) means the distance between the moving rows d rows measured longitudinally, or the clearance blade row blade row moving.

side clearance forward is generally slightly more than the rance aft.

arance.

turbine of large power the clearance between the blade rows, d in a fore and aft direction, are as follows:—

st Expansion H.P.	-	From	·242 in.	to	·263 in.	(Forward)
nd " "	-	"	·213	"	·253 "	"
rd " "	-	"	·264	"	·246 "	"
th " "	-	"	·320	"	·420 "	(Aft)
st " L.P.	-	"	·353	"	·323 "	(Forward)
nd " "	-	"	·338	"	·343 "	"
rd " "	-	"	·413	"	·330 "	"
th " "	-	"	·380	"	·380 "	"

—Dummy setting, ·025 in. or $\frac{25}{1000}$ in.

ughout any expansion the clearance usually varies, and is constant, some of the intermediate blade rows having more de clearance than the first or last blade rows.

ip Clearance.

(sured cold.)

1st Expansion	- - -	From	·043 in.	to	·041 in.
2nd "	- - -	"	·035	"	·012 "
3rd "	- - -	"	·027	"	·023 "
4th "	- - -	"	·020	"	·048 "

ll be seen from the foregoing that the tip clearance is greatest st expansion, and falls off in value at the 2nd, 3rd, and 4th ons. This difference is accounted for by the fact that the ture is highest at the 1st expansion, and falls at each on in succession; so that the required allowance for blade on due to heat becomes less proportionally.

—In turbines at present under construction blade tip clear- almost been done away with altogether, the thinned tips of es just clearing and no more.

al Column Vacuum Gauge.

ng turbine trials the vacuum is tested by means of a column ry (barometer), as shown in the sketch (12), the open end of s tube being connected up to the condenser or L.P. exhaust as required, and the other end kept closed. The

-Throughout the text of the New Sections the illustrations are referred r rotation number. Thus: "as shown in the sketch (12)" and so on for ber.

Action of Steam in Dummy Rings (H.P. Turbine).

The sketch (13) shows the approximate fall of pressure due to alternate wire drawing and expansion of the leaking steam which takes place in the dummy rings and grooves. Notice that the pressure at the back of the last ring is equal to that inside the rotor, which is the exhaust steam pressure of that turbine. In L.P. turbines the final pressure will be equal to that of the vacuum pressure of the exhaust steam.

A general idea of the action of the steam in leaking past the rings, and falling in pressure and in quantity, may be obtained from the following :—

DATA.

Dummy 7 ft. 6 in. mean diameter at grooves.
Pockets formed by rings $\frac{3}{8}$ in. by $\frac{1}{2}$ in.
Pressure in pocket behind first ring = 157 lbs. gauge.
" " before last " = 35 "

At First Pocket.

Then, $157 + 15 = 172$ lbs. absolute $= 2\cdot61$ cub. ft. volume per lb.

Note.—7 ft. 6 in. = 90 ins.

Then, Volume of pockets $= \dfrac{90 \text{ in.} \times 3\cdot1416 \times \cdot375 \times \cdot5}{1728} = \cdot0306$ cub. ft.

So that, $\cdot0306 \div 2\cdot61 = \cdot0117$ of a lb. steam present.

At Last Pocket.

$35 + 15 = 50$ lbs. absolute $= 8\cdot32$ cub. ft. volume per lb.

Then, $\cdot0306 \div 8\cdot32 = \cdot0037$ of a lb. steam present.

The weight of steam leaking past the steam rings is thus checked by the number of expansions in the pockets formed by the rings and grooves. For equal volume of pockets the *weight* of water or steam present becomes less as the pressure falls and the volume increases.

Note.—As superheated steam (100° Fahr.) is now being introduced into turbine practice, copper is the material substituted for the dummy rings in place of brass. The blades of the first three or four expansions are also composed of copper, which metal is expected to better withstand the effects of the superheat than brass.

Description of the Propelling Machinery of the Q.SS. "Reina Victoria Eugenia" (14).

Constructed by Messrs Swan, Hunter, & Wigham Richardson Ltd., Newcastle-on-Tyne.

The propelling machinery consists of two complete units of reciprocating engines and turbines, driving four screws in all.

The reciprocating engines are of the well-known four-crank triple expansion type, balanced on the Yarrow-Schlick-Tweedy principle, with cylinders 29 in., 43 in., 47 in., and 47 in. in diameter, and a stroke of 42 in. The low-pressure cylinders at each end of the engines are designed to develop less power than the two other cylinders, so that a very uniform turning moment is obtained, in addition to good balancing.

Each engine exhausts into a steam turbine through special manœuvring-valves mounted at the back of each low-pressure cylinder, and operated by levers on the reversing shaft. These valves are so arranged that the exhaust steam is passed direct to the condenser when the reversing gear is in the astern position. Thus all manœuvring is done by the reciprocating engines on the inboard shafts. Provision is made, by means of screw and hand-wheel, whereby the turbines can be cut out entirely by passing the steam directly into the condenser under all conditions.

The turbine installation comprises two low-pressure Parsons' turbines driving the wing-shafts, and designed for working ahead only. Steam enters each turbine at the forward end through two 18½-in. diameter pipes—*i.e.*, one from each low-pressure cylinder—with a suitable strainer in each branch, the admission pressure being about 10 lbs. per sq. in. absolute, with a maximum steam consumption of 135000 lbs. per hour. The exhaust branch, having a cross-sectional area of 30 sq. ft., is designed for a condenser pressure of 1 lb. absolute. The rotor drum is parallel, 68 in. in diameter, and carries thirty rows of blades. The dummy at the forward or steam end is fitted with labyrinth-packing strips of the radial type—ten strips in the rotor and ten in the casing. The diameter of the dummy is such that the total axial steam thrust on the rotor substantially balances the propellor thrust. The rotor shaft glands are of Parsons' combined labyrinth and ring type, each being fitted at the inner end with ten rows of moving, and ten rows of fixed rings, and with four Ramsbottom rings at the outer end. The glands are so designed as to be easily overhauled without lifting the turbine-cover. The rotor journals at each end are 12 in. in diameter, working in white metal bearings, each 23½ in. long. A rotor adjusting-block is fitted at the forward end for the purpose of adjusting the axial position of the rotor, and contains ten rings, which bear upon the faces of corresponding collars on the rotor shaft. Lubricating oil is supplied under pressure to the bearings and adjusting-blocks by a duplex oil-pump. The oil draining from the bearings, etc., is collected in a tank and

to the three screws, generally applied with the com

Th

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500 gal
e air-pum
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The oil draining from the bearings, etc., is collected in a tank and

cooled before being discharged again to t
turbines is illustrated on opposite page.

The condensers are of the "Uniflux'
cooling surface of 5100 sq. ft., designed for
an average sea temperature of 75°. The
supplied by Messrs H. Watson & Co. hav
charging 8500 gallons per minute against 2

The air-pumps are of Weir's latest "
24 in. by 17 in. each. There are further in
feed-pumps, with cylinders 13½ in. by 10
evaporators, two 90-in. Howden fans, t
12-ton distiller, a ballast donkey, two gener
smaller pumps, a refrigerating plant, and a
ing machine, which has already been illu
"Engineering." The exhaust steam from
for heating up the feed-water, a "Neptune
stalled for this purpose.

There are seven single-ended boilers
diameter and 11 ft. 6 in. in length, workin
forced draught. Each boiler has three larg
grate surface of 480 sq. ft.: the total heati
and the working pressure 180 lbs. per sq. in

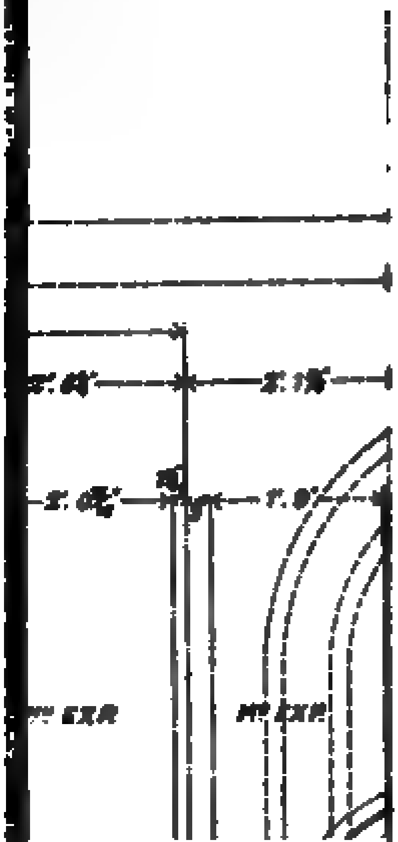
On trial the ship half laden was requir
eight consecutive hours, and when fully lac
for twenty-four consecutive hours. The a
given in the table on p. 372, and show tha
a speed of 18·12 knots was obtained in adv
twenty-four hours' trial the speed obtained
weather conditions and in deep sea, it is c
would have been secured, and we are info
deep sea and fine weather, the speed obta
excess of that realised at the trials carried c

The steam consumption was measure
nozzles regularly employed by the build
figures given in the table include made-up
Progressive runs over the measured mile a
east coast, were made during each trial i
speeds obtained on the respective trials
recorded on a double run was 18·6 knots
75 ft. deep), as, owing to the foggy weather
be seen a small distance from the shore.

A set of indicator cards taken during
shown, and the results are of special interes
pulsive efficiency indicated by the low co
should be mentioned in connection herew
made by the builders some years ago w
have induced them to choose the four-screw
to the three screws, generally applied with

Results of Trials of "Reina Victoria-Eugenia."

	8 hours, half-loaded	24 hours, fully loaded
	February 7, 1913	February 15 and 16, 1913
	19 ft. 10 in.	24 ft. 8½ in.
ded)	10181 tons	13229 tons
	0 ft. to 6 ft.	Smooth
	18 knots ahead to 30 knots starboard bow, 30 knots port quarter to 40 knots port beam	Varying between 0 and 10 knots
	29·7 to 29·5 in.	30·4 in.
in each	2 × 71 knots	4 × 89 knots
	130 ft.	150 ft.
	18·12 knots	16·10 knots
es	112·6 r.p.m.	102·9 r.p.m.
	1 per cent. slip	4 per cent. slip
	481 r.p.m.	395 r.p.m.
	20 per cent slip	13 per cent. slip
rer	7340	5760
	3500	2157
	10840	7910
	257	294
st.	170 lbs. per sq. in.	170·5 lbs. per sq. in.
	7·8 lbs. per sq. in. absolute	5·4 lbs. per sq. in. absolute
	0·5 lbs. per sq. in. absolute	0·56 lbs. per sq. in. absolute
	43 deg. Fahr.	45 degs. Fahr.
irge	70 ,,	64 ,,
ell	70 ,,	62 ,,
	...	183 ,,
iption for	114000 lbs. per hour	85000 lbs. per hour
nption for	...	14000 ,, ,,
pits	0·4 in water	0·35 in. water
by calori-	14200 B.Th.U.	14400 B.Th.U.
ır	...	3 per cent.



SECTION VII.

IMPULSE TURBINES, COMBINED IMPULSE AND REACTION TURBINES, AND GEARED-DOWN TURBINES.

Principle of Impulse.

If the impulse wheel blade speed is equal to the steam speed the blades will extract no energy from the steam jet, but if the blade speed is *less* than the steam speed then energy will be extracted, proportional to the respective speeds of jet and blades.

Assume the following:—

Steam speed 2000 ft. per sec., blade speed, 1000 ft. per sec.

Then, Relative speed of steam = 2000 — 1000 = 1000 ft.

Therefore energy equal to 1000 ft. velocity will be extracted from the steam in striking the moving blade rows.

Again, Relative exit speed of steam = 1000 ft. and, 1000 — 1000 = 0 ft. final velocity of steam.

From the above it will be evident that for a simple impulse turbine, fitted with one moving blade row, the blade speed should be equal to one-half of the steam speed, as with this condition obtaining, the steam will part with the whole of its kinetic energy, and will leave the blades at zero velocity.

In actual impulse turbines the steam speed is not much less than 2000 ft. per sec., but the blade speed is usually under 200 ft. per sec.; as, however, a number of moving blade rows and fixed blade rows are fitted, the surplus velocity or energy still remaining in the steam, after passing through the first moving blade rows, is used up in the other blade rows so that finally the whole of the velocity is absorbed and the exhaust steam possesses only a low velocity which is known as a "carry over" loss. Theoretically, the steam should possess no velocity after leaving the last moving blade row of any impulse stage, as such velocity represents kinetic energy unused and therefore wasted.

Steam Velocity Increase through Impulse Nozzles.

When the pressure at the throat of a nozzle is reduced to ·58 of the initial pressure entering the nozzle, the limit of steam flow is reached, and no matter how much the pressure is reduced below this the quantity flowing through into the turbine wheel blades remains the same. If the throat pressure exceeds ·58 of the initial

pressure the quantity discharged is checked. Nozzles in turbines are generally of the convergent-divergent type, as shown in the sketch, the taper beyond the throat being about 1 in 12 to 1 in 26.

The heat drop due to fall of pressure is converted into jet velocity or kinetic energy, and the acquired velocity may be calculated as follows, neglecting practical losses such as friction, etc.

Rule, $V = \sqrt{64 \cdot 4 \times 778 \times \text{Heat Drop in B.T.U.}}$

Example.—**Pressure at nozzle box, 180 lbs. gauge, dryness = 1, pressure at impulse wheel, 60 lbs. gauge dryness = ·92 (assumed).**

$$180 + 15 = 195 \text{ lbs. absolute} = 379{\cdot}7 \text{ Temperature Fahr.}$$
$$60 + 15 = 75 \quad " \quad " \quad = 307{\cdot}5 \quad " \quad "$$
$$195 \text{ lbs.} = 846{\cdot}5 \text{ B.T.U. Latent Heat (from Tables).}$$
$$75 \quad " \quad = 897{\cdot}5 \quad " \quad " \quad "$$
$$379{\cdot}7 + 461 = 840{\cdot}7 \text{ Absolute Temperature.}$$
$$307{\cdot}5 + 461 = 768{\cdot}5 \quad " \quad "$$

Then, $\text{Heat Drop} = 846{\cdot}5 \times 1 - 897{\cdot}5 \times {\cdot}92 + 840{\cdot}7 - 768{\cdot}5.$

" " $= 846{\cdot}5 - 825{\cdot}7 + 840{\cdot}7 - 768{\cdot}5.$

" " $= 1687{\cdot}2 - 1594{\cdot}2.$

" " $= 93$ B.T.U.

So that, $\text{Velocity} = \sqrt{64{\cdot}4 \times 778 \times 93} = 2158$ ft. per sec.

Degree of Reaction and Blade Speed.

If the so-called degree of reaction of a single impulse wheel is equal to ·5 then the blade speed would require to be equal to $2158 \div 2 = 1079$ ft. per sec., and assuming a mean diameter of, say, 4 ft. The revolutions required per minute would be—

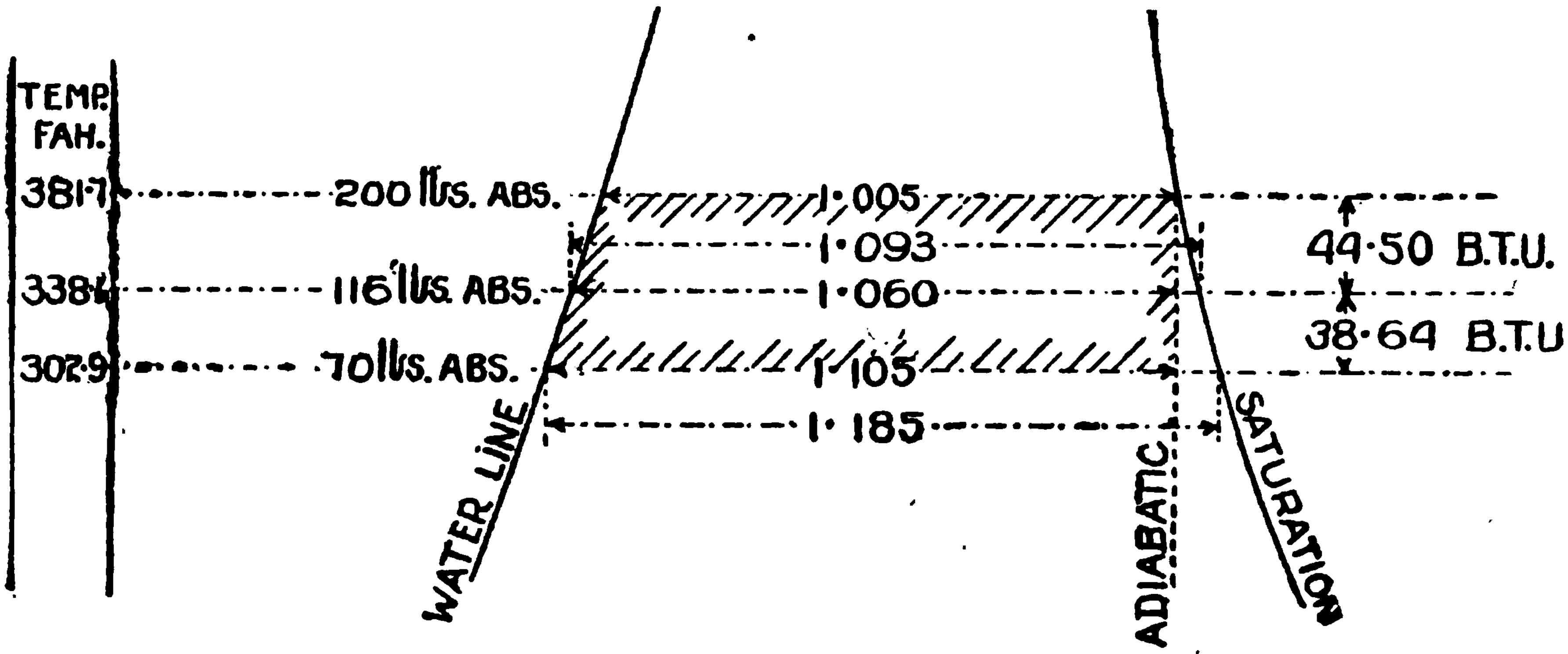
$$\frac{1079 \times 60}{4 \times 3{\cdot}1416} = 5151.$$

By increasing the number of blade rows per stage the blade speed can, however, be much reduced as each moving row extracts velocity from the steam.

Data—

Steam at nozzle box, 185 lbs. gauge = 200 lbs. absolute.
Steam at impulse stage, 55 gauge = 70 lbs. absolute.
Condition of steam, dry.
Nozzle efficiency, 90 per cent.
Adiabatic expansion, assumed.

Then, Throat pressure = $200 \times {\cdot}58 = 116$ lbs. absolute.



No. 18.—Entropy Diagram.

Calculations—

Referring to Entropy Diagram—(18).

Between pressure limits of 200 lbs. and 116 lbs. mean entropy $= \frac{1\cdot005 + 1\cdot060}{2} = 1\cdot0325.$

Temperature difference $= 381\cdot7 - 338\cdot6 = 43\cdot1.$

Then, Drop in B.T.U. $= 1\cdot0325 \times 43\cdot1 = 44\cdot5$ **B.T.U.**

Steam velocity at throat $= \sqrt{44\cdot5 \times 64.4 \times 778 \times \cdot90} = 1416$ **feet per second.**

Dryness $= 1\cdot060 \div 1\cdot093 = \cdot969.$

Referring to Entropy Diagram—(18).

Between the pressure limits of 200 lbs. and 70 lbs. the mean entropy $= \frac{1\cdot005 + 1\cdot105}{2} = 1\cdot055.$

Temperature difference $= 381\cdot7 - 302\cdot9 = 78\cdot8.$

Then, Drop in B.T.U. $= 1\cdot055 \times 78\cdot8 = 83\cdot13.$

Steam velocity at mouth $= \sqrt{83\cdot13 \times 64\cdot4 \times 778 \times \cdot90} = 1936$ **feet per second.**

Dryness $= 1\cdot105 \div 1\cdot185 = \cdot932.$

Heat Drop and Steam Velocity.

If a pressure drop or heat drop takes place, work of some kind is being done by the steam, either (*a*) useful work such as change of pressure into velocity, work rotation of the shaft, etc.; or (*b*) what may be termed useless work such as, for example, overcoming friction, making up for heat losses, shock on turbine blades, etc.

A pressure fall results in increase of steam velocity, in the case of the Parson's reaction blading, but if the pressure remains constant then the velocity may decrease as in the case of Curtis' or Parson's impulse wheel blading, work being then extracted from the steam within the moving blade rows.

Pressure Compounding.

Impulse turbines are compounded for pressure by increasing the number of nozzle stages (56), and thus dividing the pressure drop into steps.

Velocity Compounding.

Impulse and reaction turbines are compounded for velocity by increasing the number of blade row stages or expansions. Impulse

and reaction turbines can therefore be compounded for either velocity or pressure, or for both velocity and pressure (19).

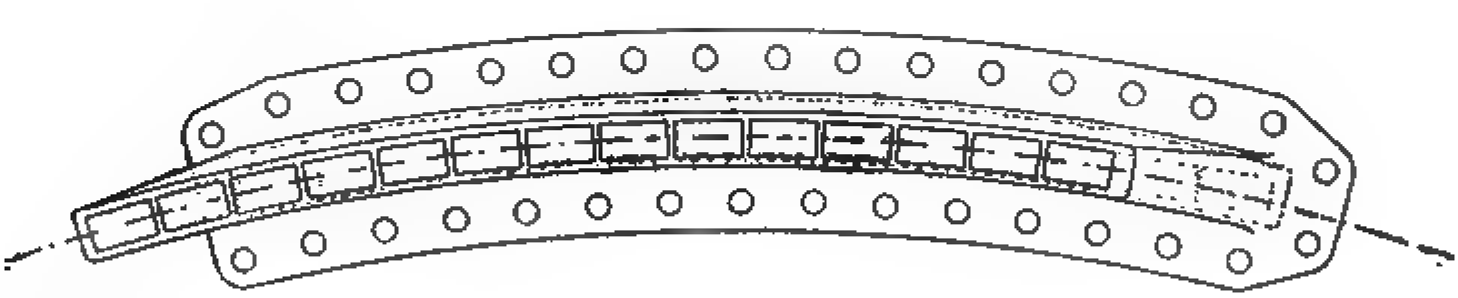
In the impulse stage expanding nozzles the steam falls in pressure but increases in velocity; in the wheel blades the pressure remains practically constant but the absolute velocity falls, as work is then extracted from the steam by way of its jet velocity energy.

SECTION THRO' ASTERN TURBINE
IMPULSE WHEEL

No. 19.—Impulse and Reaction Turbine.

The impulse stage shown above consists of 3 rows of moving blades, and 2 rows of fixed blades. Notice that the blades of the first reaction expansion are much smaller than the last blades of the impulse wheel.

On comparing impulse blading with the Parson's reaction blading it will be noticed that the impulse blades give equal openings for steam flow at both the admission and exit edges, whereas in the reaction blading the exit openings are less than the admission

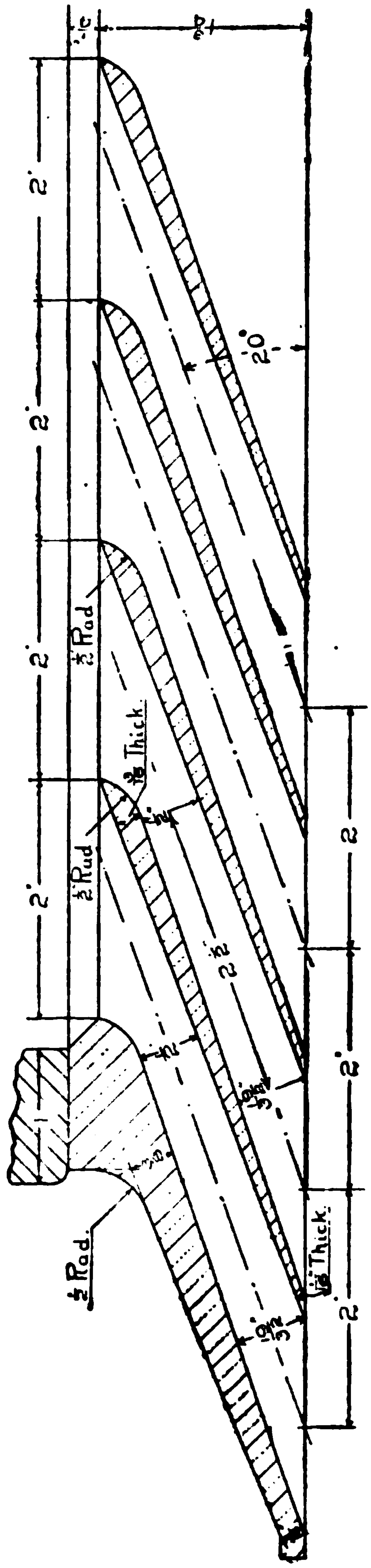


No. 20.—Nozzle Openings of Parson's Turbine.

openings, the openings ratio for normal blades being as 3 is to 1; thus difference in area resulting in increase in steam velocity.

Briefly, then,

Nozzle Stages represent pressure compounding.
Blade Rows represent velocity compounding.



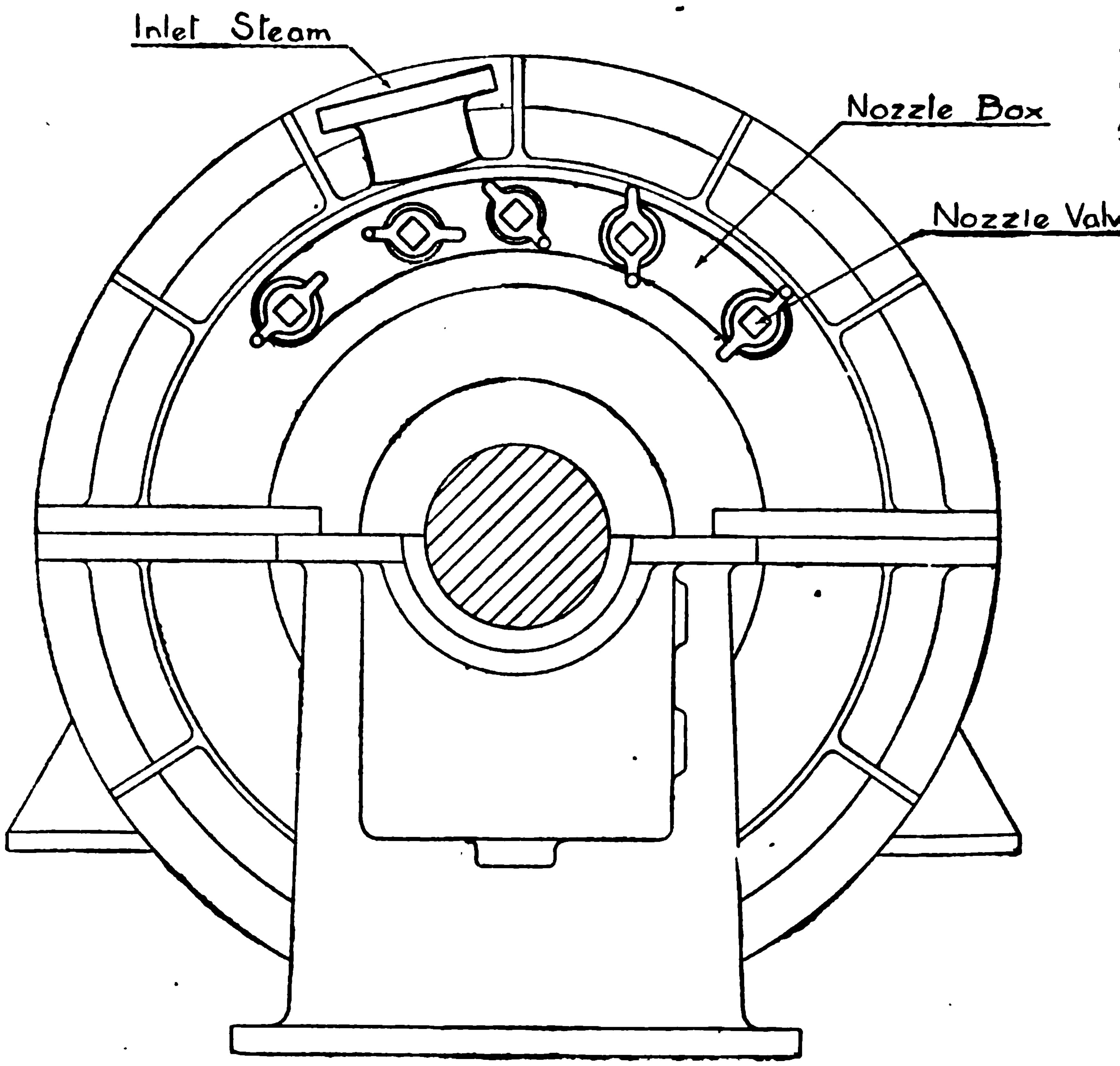
No. 21.—Nozzles of Parson's Combined Impulse and Reaction Turbine.

The taper is 1 in 26.

tea

Combined Impulse and Reaction Blading.

Parson's turbines are now generally fitted at the initial end with a wheel and a set of impulse blading similar to that of the Curtis turbine, the remainder of the blading being of the reaction pattern. In some cases the astern turbines only are supplied with impulse blading so as to allow of quick picking up of power when running



No. 23.—End View of Turbine Casing.
Showing Nozzle Control Valves.

astern. In other cases both reverse and ahead turbines are fitted with impulse blading and wheel at the first stage expansion, the other expansions being of the usual reaction type blading.

Impulse blading has the advantage of allowing the full power to be developed more quickly than is possible with reaction blading alone, a point of considerable value in naval work. The steam is

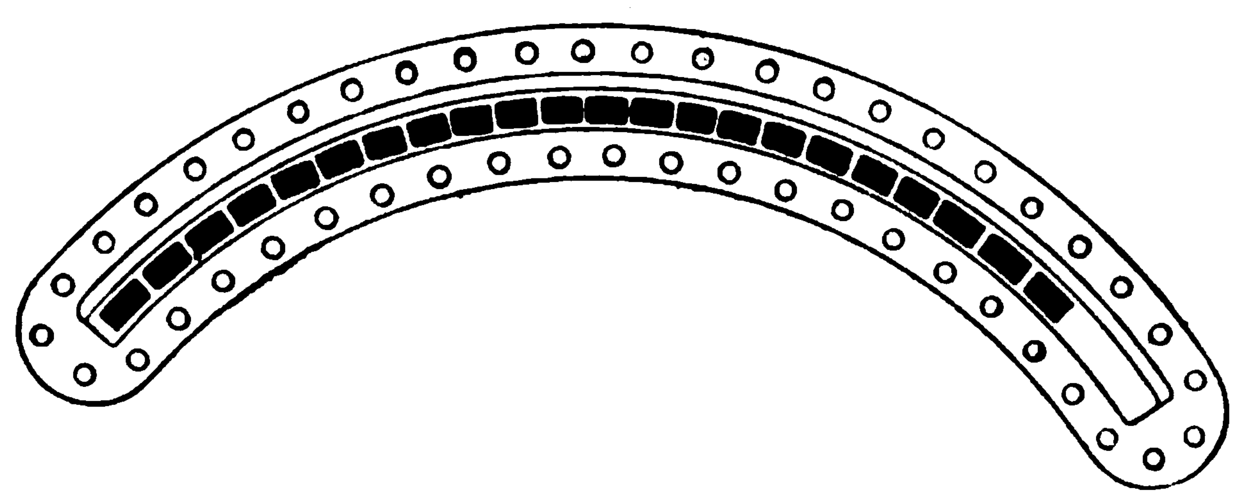
Pressures.

After the steam passes through the impulse nozzles and blading a considerable pressure drop takes place, and a corresponding increase in steam velocity, and this change is allowed for in the reaction blade expansions which are designed accordingly.

admitted to the impulse wheel blades through expanding nozzles, the quantity being controlled by means of hand valves, each one having a known area of opening equal to a certain number of nozzles. These area values are marked on the hand wheels of the controlling valves.

Nozzle Box.

The nozzle plate forms a segment only of the turbine casing at the top and does not, it should be noted, extend right round; the wheel is, of course, complete and the impulse blades of the wheel also complete the circle. It should, however, also be noted, that the impulse guide blades which are fixed in the casing shell extend round the wheel circle for the **top half only**. The nozzle box is bolted on to the end of the turbine casing and is fitted with a stop valve which admits steam to the box, also with a number of hand controlling valves (24) which allow the steam to pass from the

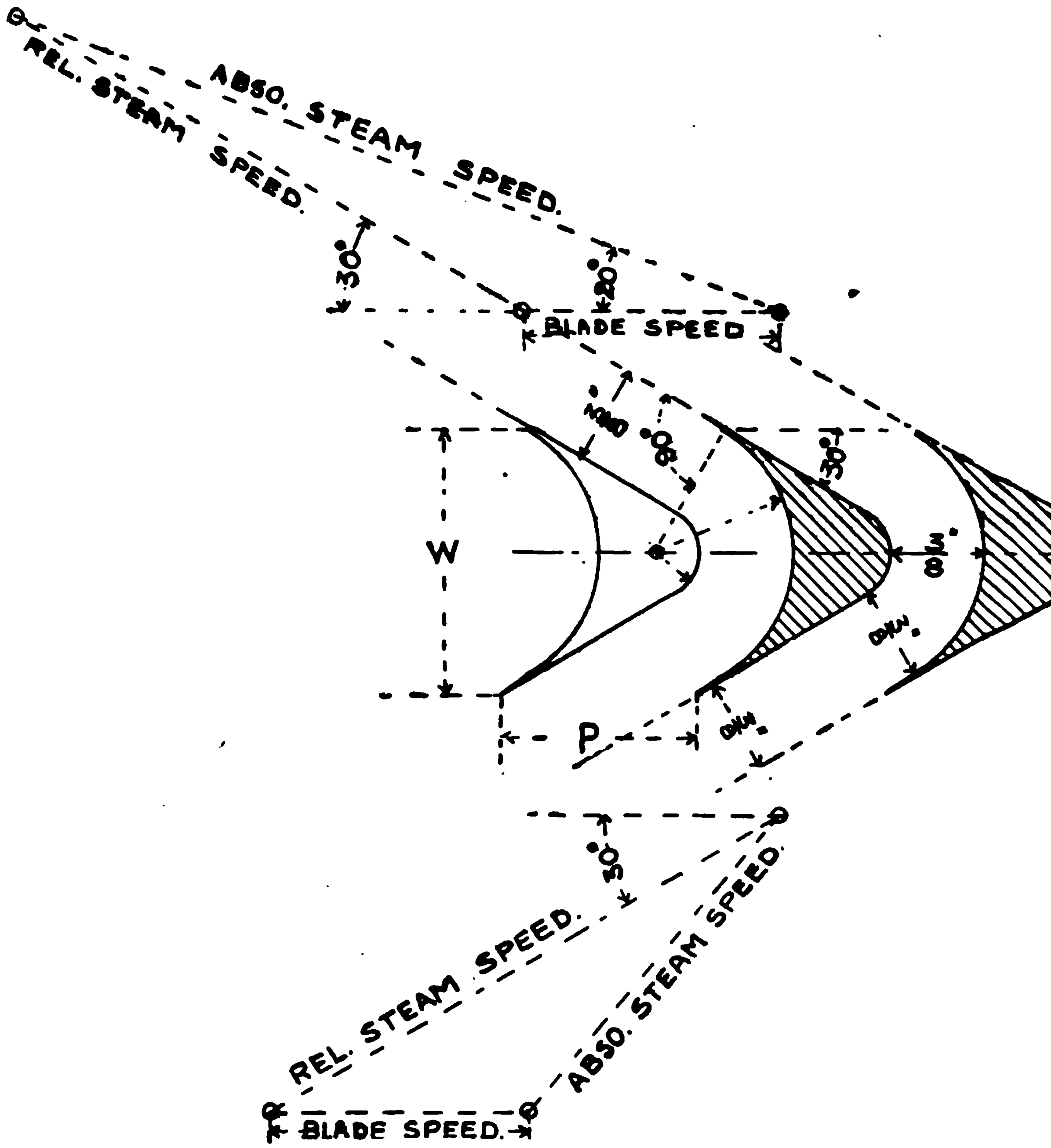


No. 24.—Nozzle Openings.

nozzle box to the nozzles, and so to the impulse blades of the wheel. After expanding through the impulse nozzles and blading, and dropping in pressure the steam enters the first expansion of the reaction blading which are very much less in height than those of the impulse wheel. This will easily be understood, when it is remembered that the nozzle openings only extend round the casing for a very small part of the circumference, whereas the reaction blades complete the circle, and in this way balance up the area of steam flow at the reduced pressure. For given powers and speeds certain of the control valves are opened, and at full power all may be opened up to give full steam flow through the nozzle openings. Economy is, therefore, obtained at all powers and the necessity for cruising turbines to a great extent eliminated.

Pressures.

After the steam passes through the impulse nozzles and blading a considerable pressure drop takes place, and a corresponding increase in steam velocity, and this change is allowed for in the reaction blade expansions which are designed accordingly.



No. 25.—Diagram of Velocities for Impulse Blading.

The blades shown with dimensions are of the Parson's type.

Note. **The pitch, P of the blades is usually equal to about ·75 of the width, W.**

Advantage of Impulse Blading.

The impulse blading, being much heavier and stronger than the reaction blading, allows of greater steam flow when starting up without risk of damage to blades, and results in a reduced steam pressure admission to the first stage of the reaction blading of the turbine, which is thus less subject to shock or vibration when steam is first turned on for either ahead or astern. It will thus be evident that the maximum steam pressures are exerted on the impulse

in Reaction Blades.—Fall of pressure, fall of temperature, and increase of velocity with decrease of dryness.

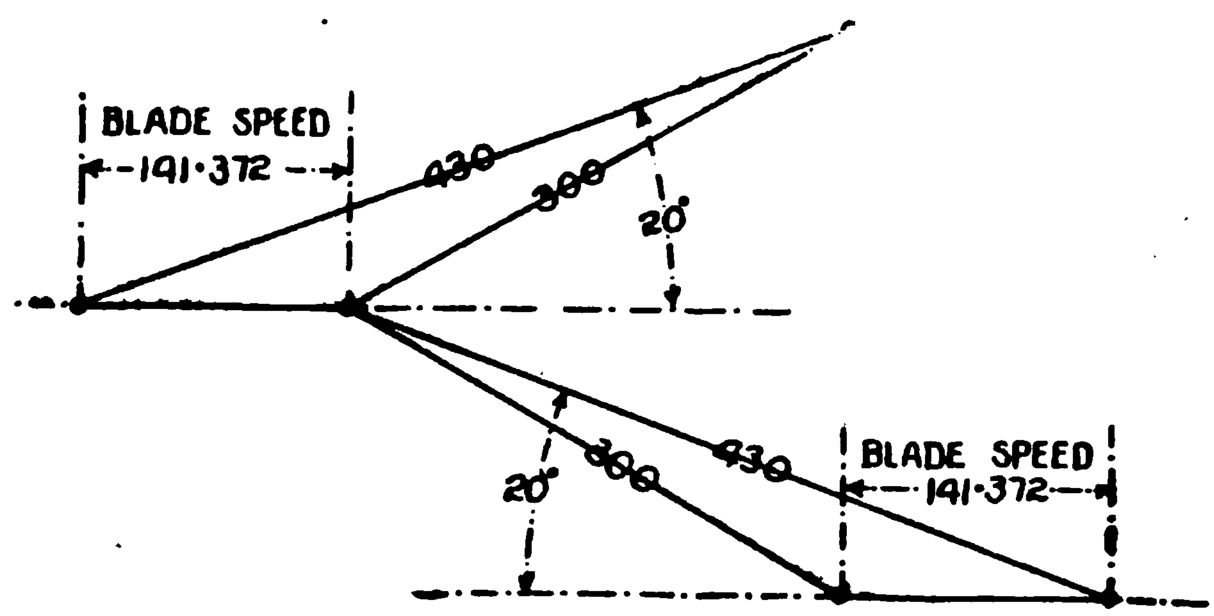
It should be noted that in the nozzles the expansion of the steam results in fall of pressure accompanied by increase of jet

pressure admission to the first stage of the reaction blading of the turbine, which is thus less subject to shock or vibration when steam is first turned on for either ahead or astern. It will thus be evident that the maximum steam pressures are exerted on the impulse

blading only, the reaction blading receiving the steam at greatly reduced pressure.

Cruising Nozzles.

For low cruising speeds only a few of the control valves are



No. 26.—Velocity Diagram of Reaction Blading.

The velocity triangles are equal for admission and exit.

opened and these admit steam through a fixed number of special cruising nozzles; if the power and speed have to be increased, additional valves are opened out which thus uncover more of the nozzles to steam flow at the same pressure.

For economy at cruising speeds some engineers advocate throttling the steam on its way to the turbine, by means of the stop valves, to reduce the pressure at the nozzle box. This practice, it will be noted, is opposed to that of reciprocating engines where economy is effected by carrying full steam at the H.P. chest and linking up the H.P. valve gear, and is theoretically wrong.

Note.—In place of fitting cruising nozzles as described, it is now usual to fit special cruising turbines in cruisers, battleships, and torpedo boats (see p. 457).

Action of the Steam.

The changes in pressure, velocity, and temperature of the steam, when flowing through the nozzles and blades of a combined impulse and reaction turbine, are approximately as follows:—

In Nozzles.—Fall of pressure, increase of velocity, and fall of temperature with decrease of dryness.

In Impulse Wheel Blades.—Constant pressure, decrease of velocity, and decrease of temperature with decrease of dryness.

In Reaction Blades.—Fall of pressure, fall of temperature, and increase of velocity with decrease of dryness.

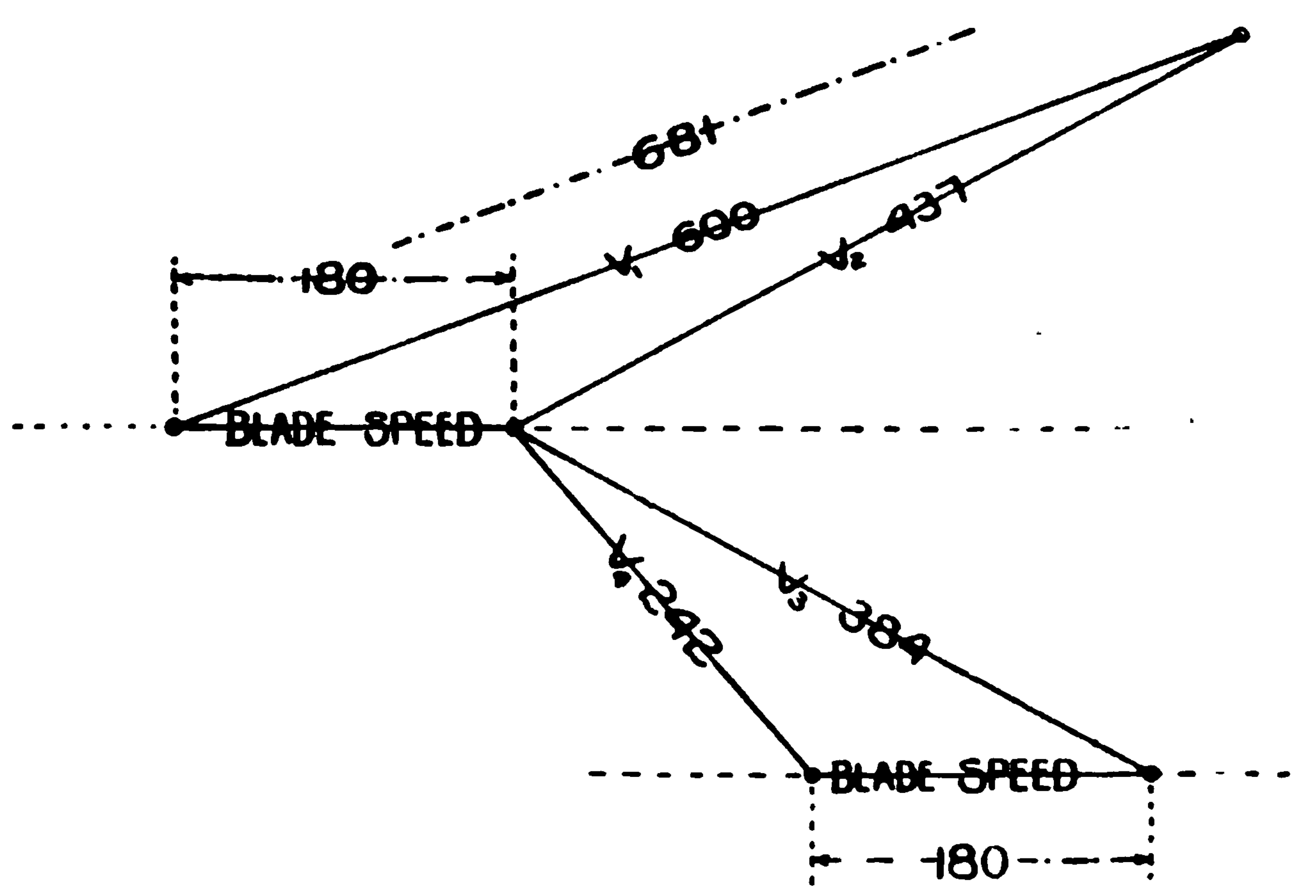
It should be noted that in the nozzles the expansion of the steam results in fall of pressure accompanied by increase of jet

velocity and therefore work energy. In the blades of the impulse wheel part of this energy is extracted in the form of effective work of rotation, so that if the pressure remains constant the velocity is reduced in proportion to the heat taken out by way of work done.

On examining the impulse blading it will be seen that the steam flow openings are practically parallel, the exit openings being equal to the admission openings, whereas in reaction blading the exit openings are always less than the admission openings.

Guide and Moving Blades.

As energy is taken out of the steam during its flow through the



No. 27.—Velocity Triangles of Impulse Blading.

Assuming a velocity loss of 12 per cent. in nozzles and blades—

Then, $681 \times \cdot 88 = 600$ **feet actual velocity.**

$437 \times \cdot 88 = 384$ „ „

Note.—$681 =$ **calculated velocity.**

$$100 - 12 = 88 \text{ or } \frac{88}{100} \text{ and } \frac{88}{100} = \cdot 88 \text{ efficiency.}$$

moving blades, by work done on the rotor, if the pressure remains the same then the speed of the steam will fall; but as in the guide blades no external work is done, then the heat energy contained results in constant pressure and velocity being maintained.

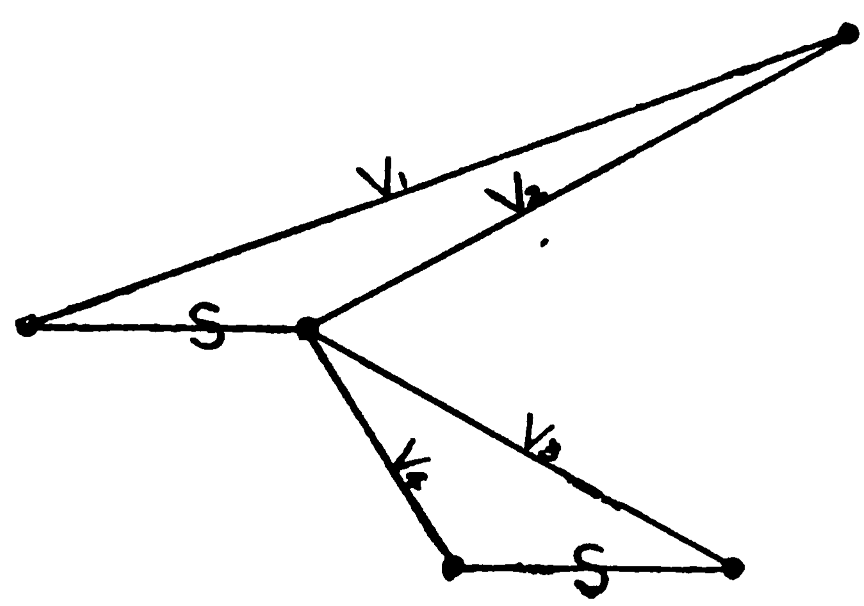
Each pound of steam passing through a turbine contains a known number of available heat units, which can be utilised in either the doing of external work (moving blades), or in the increase of the velocity of the pound of steam (nozzles).

It will thus be evident that if work is done on the moving blades heat is extracted, or if work is done on the steam itself to increase its velocity heat is also extracted. Both of these conditions maintain in

a turbine, as the steam flows alternately through the moving blades and guide blades. The guide blades of reaction turbines are to all intents and purposes miniature nozzles and should be considered as such when studying the action of, and changes in condition of, steam flow through the expansions of a reaction turbine.

The heat of the steam is either expended in doing useful work, or in accelerating the velocity of the steam, or in both at one time.

Work can only be done at the expense of heat, and the work done may either be external work of shaft rotation or, the internal work of steam velocity acceleration as before stated. In a turbine the work is done by the energy of jet velocity, and in a reciprocating engine the work is done by static pressure, but in both cases heat is, of course, given up in exact proportion to the work done, each 778 foot-lbs. of work being equivalent to 1 B.T.U. of heat given up.



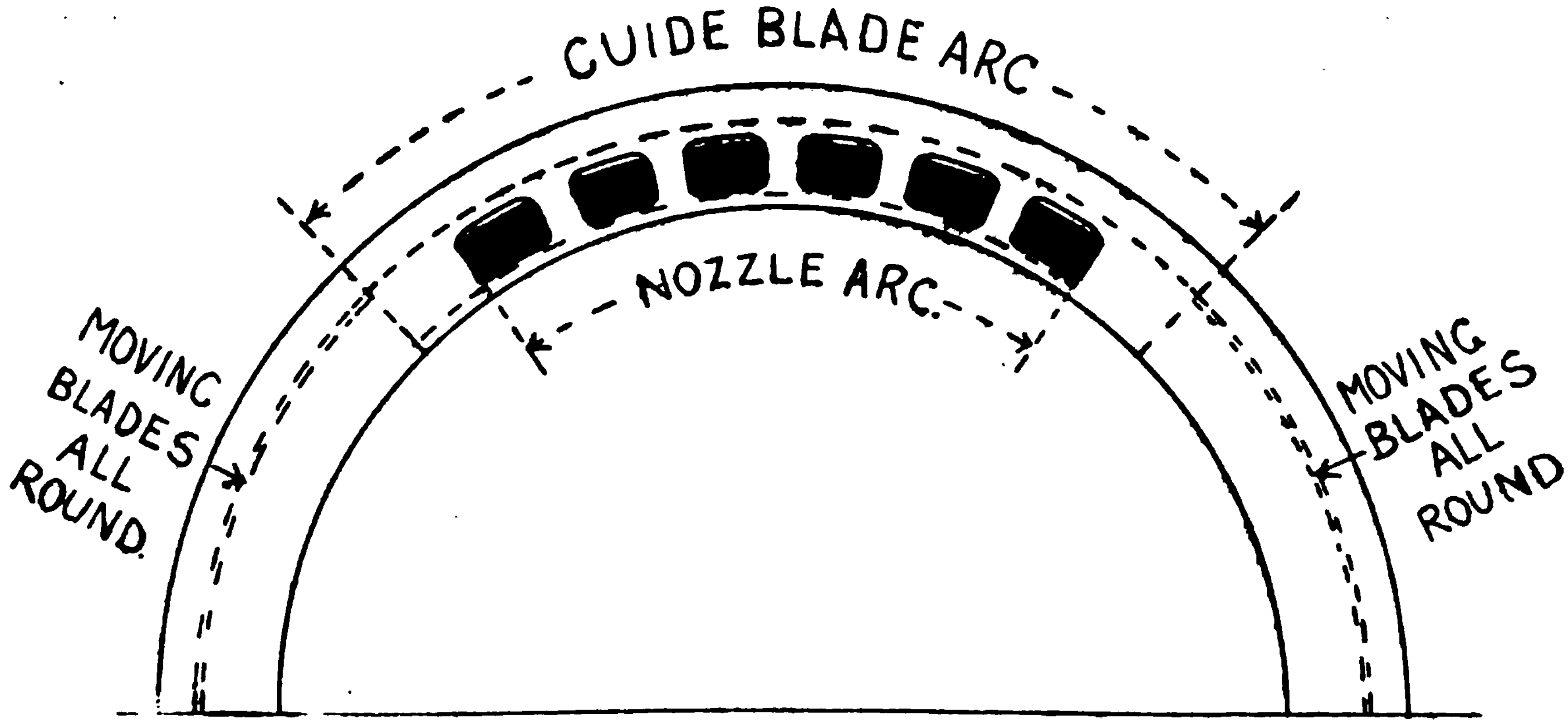
No. 28.—Velocity Triangles of Impulse Blading.

Total kinetic energy contained in 1 lb steam $= \frac{V_1^2}{64 \cdot 4} =$ **foot lbs.**

Kinetic energy extracted from each 1 lb. steam in moving blades $= \frac{V_1^2 - V_4^2}{64 \cdot 4} =$ **foot lbs.**

Blades in Use.

As the steam enters through the nozzles, which only occupy a segmental arc of the turbine (29), only a certain number of the moving blades are taking steam, but as the steam travels through successive blade rows "spreading" occurs, and more blades of each row are then taking steam. On passing through the last moving blades of the impulse wheel the steam falls slightly in pressure to fill up the clearance space between the bucket wheel of the impulse section and the first row of reaction blades, and as the rotor drum is much smaller in diameter than the impulse wheel the steam will then enter the first expansion at practically all points of the circumference, and continue doing so right through the remainder of the reaction blading.



No. 29.—Nozzle and Guide Blade Arcs

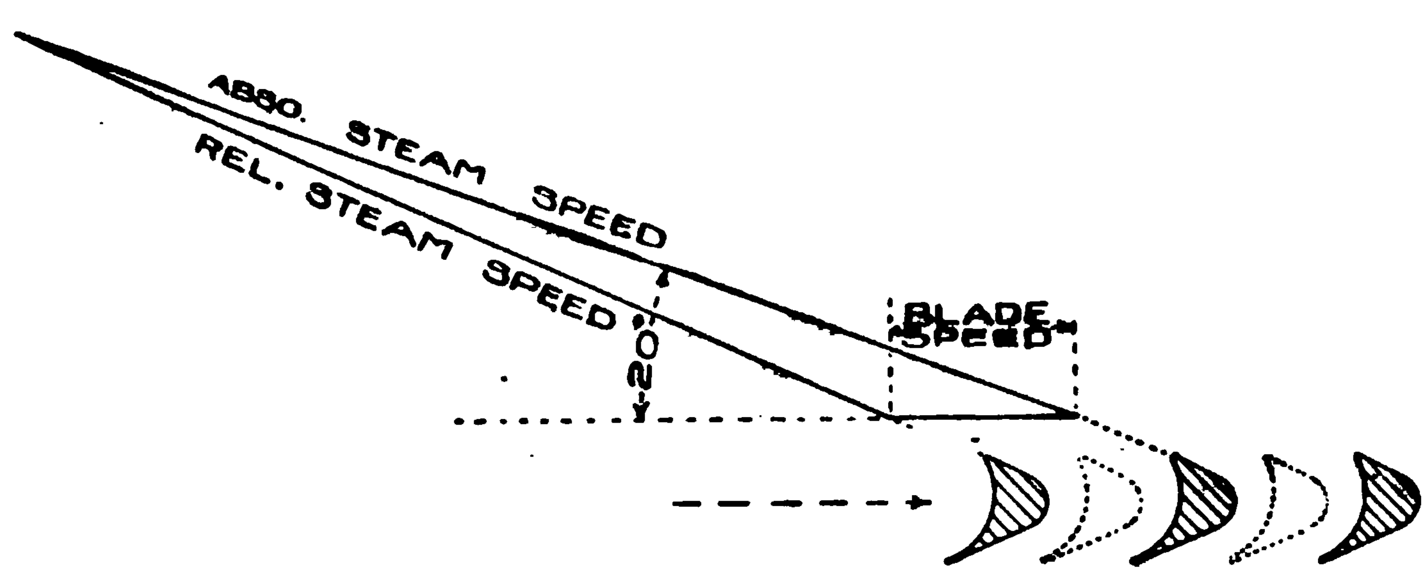
Moving Blades.

At admission edges the *absolute* velocity of the steam is in excess of the relative velocity.

At exit edges the *relative* velocity of the steam is in excess of the absolute velocity.

Fixed Blades.

At both admission and exit edges the steam velocity (relative) is the same (neglecting friction losses).



No. 29a.—Diagram showing Resultant Effect of Combined Steam Speed and Blade Speed.

Assuming the blade speed to be zero, the steam jet would strike blade A_1, but as the blades are moving with a given velocity to the right, then the steam jet actually strikes blade A_3; meantime blade A_1 has moved to the position of A_2: Therefore, the *relative* speed of the steam is equal to the shorter side (left) of the velocity triangle. If the blades were kept stationary the steam speed would be absolute and equal to the longest side of the triangle, but as the blades are moving, then, relatively, the steam speed is less.

Effects of Blade Friction.

The steam velocities are reduced by blade friction losses, the effect of which is to alter the velocity triangles, and, therefore, the blade angles at admission and exit edges; taking the losses as 10 per cent., then the steam efficiency through blades = 90 per cent.

Ratio of Velocity to Heat Drop.

It should be noted that the heat drop loss due to loss of velocity varies as the velocity squared.

If, therefore, the loss of velocity due to blade friction is, say, 12 per cent.,

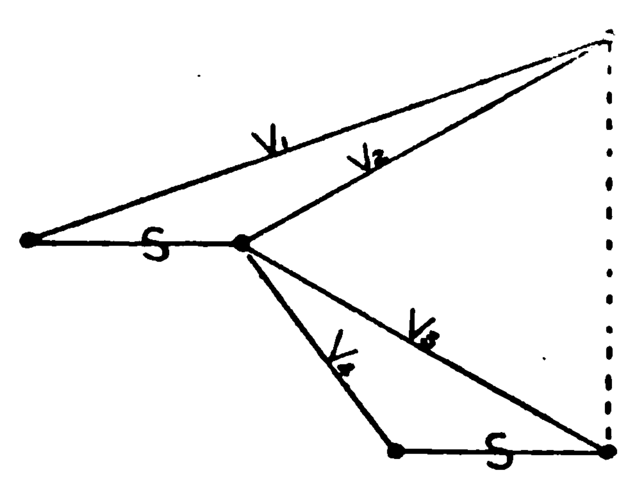
Then, $\quad$ Heat loss in foot-lbs. $= \dfrac{12^2}{64} = 22\cdot5$ per cent.

Carry-Over Loss.

In the nozzles the drop of heat due to fall of pressure is transformed into steam velocity or kinetic energy, and, having obtained active energy in this manner, the function of the blade rows is to use up as much of this energy as possible, so that when the exhaust steam passes away to the condenser its velocity will be reduced to zero, or as nearly so as practical considerations will allow of. Whatever velocity is still left in the terminal steam represents direct loss for that stage and is called the "carry-over loss."

This carry-over loss is represented in the velocity triangle by the absolute velocity of the steam (V_4) leaving the last moving blade row of the last stage.

Friction Loss.



No. 30.—Velocity Diagram, Neglecting Friction Loss.

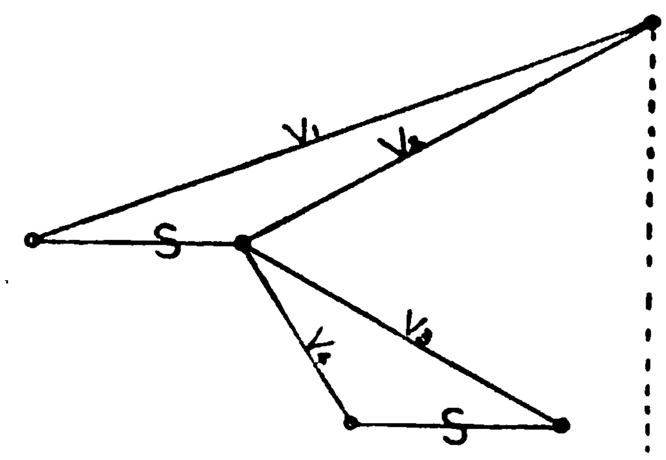
V_1 = Absolute admission steam velocity.
V_2 = Relative ,, ,, ,,
V_3 = ,, exit steam velocity.
V_4 = Absolute ,, ,, ,,
S = Blade velocity.

Friction Loss Neglected.

If friction is neglected then V_2 and V_3 are equal, so that,

$$\text{Foot-lbs.} = \frac{V_2^2 - V_3^2}{64\cdot4} = 0.$$

Friction Loss Allowed for.



No. 31.—Velocity Diagram, Allowing for Friction Loss.

When friction is allowed for V_3 is less than V_2, so that—

$$\text{Foot-lbs.} = \frac{V_2^2 - V_3^2}{64 \cdot 4} = \text{Foot-lbs. friction loss.}$$

This loss is usually taken as ranging from 6 to 12 per cent. in practice.

Force Exerted by Steam on Blades.

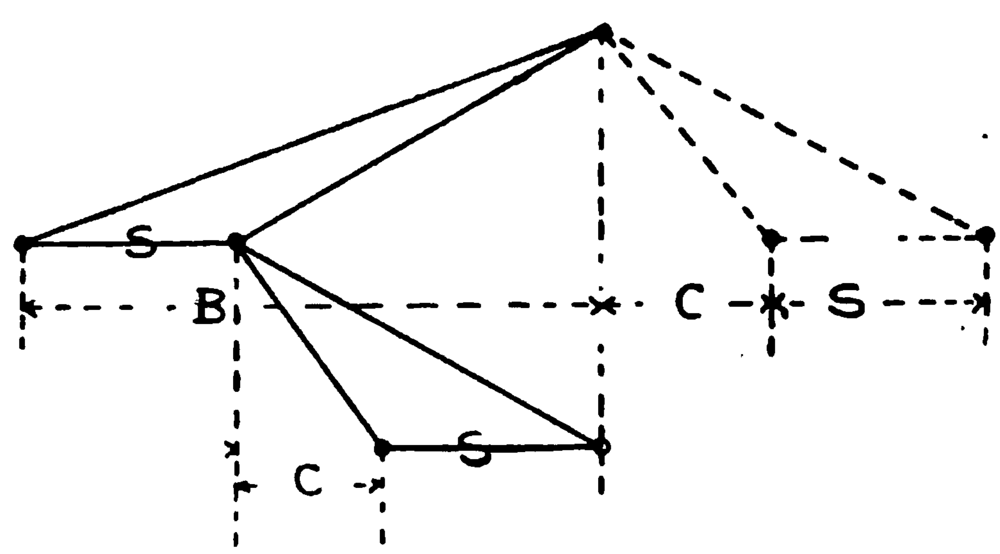
$$\text{Force in lbs.} = \frac{W \times \text{Change in velocity.}}{32 \cdot 2}$$

Where, W = Weight of Steam (1 lb.).
Where, 32·2 = Gravity acceleration constant.

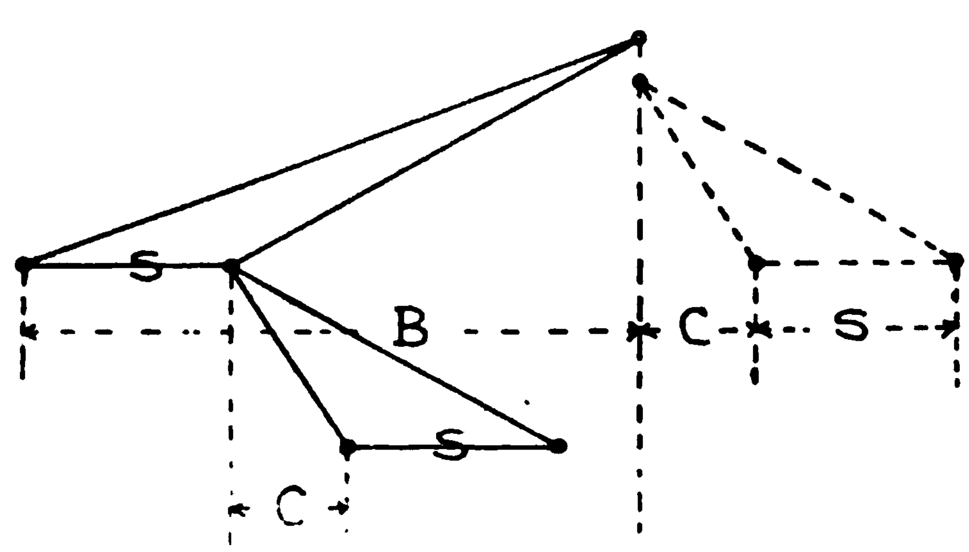
Change in Velocity.

This is determined from the velocity triangles as follows:—

Example 1.—



No. 32.—Velocity Diagram, Neglecting Friction Loss.

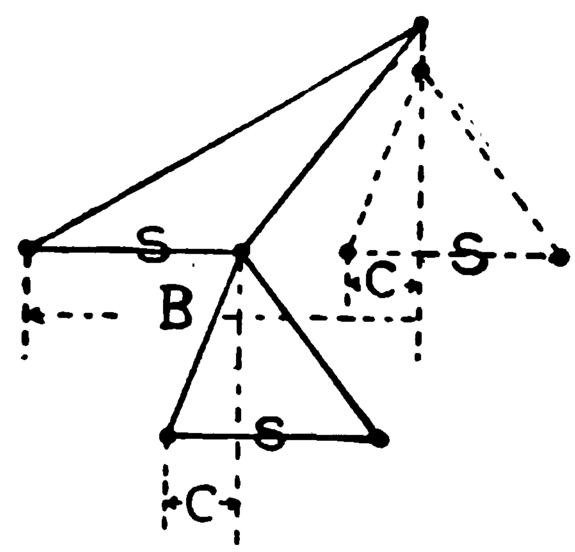


No. 33.—Velocity Diagram, Allowing for Friction Loss.
S = Blade Speed.

Convert the velocity triangles into right angled triangles as shown,

Then, **Change in Velocity = B + C.**

Example 2.

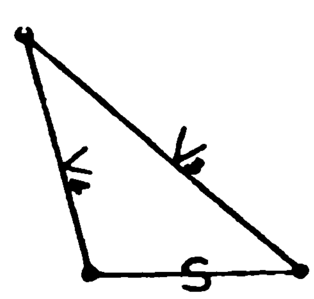


No. 34.—Velocity Diagram, with Negative Velocity.

As before, convert the velocity triangles into right-angled triangles, but as C is now in the opposite direction,

Then, **Change in Velocity = B − C.**

Carry-Over Loss.



No. 35.—Carry-over Loss.

V_3 = **Relative steam speed.**
V_4 = **Absolute „ „ (carry-over loss).**
S = **Blade speed.**

If the above triangle represents the velocity of the steam leaving the last moving blade row of a turbine in passing to the condenser,

Then, V_4 = **Carry-over loss in velocity.**

$$\text{Loss in B.T.U.} = \frac{V_4^2}{64{\cdot}4 \times 778}.$$

It should be observed that the carry-over from one stage can be utilised to some extent in the following stage.

Increase of Entropy due to Blade Friction.

$$\text{Entropy Increase} = \frac{\text{B.T.U. due to friction}}{\text{Steam Temperature} + 461}.$$

Example.—By calculation it is found that the total energy used up in overcoming blade friction = 55000 foot-lbs. in one stage, the terminal temperature of the steam being 300°. Find the increase of entropy available for the following stage.

Then, $$\text{B.T.U.} = \frac{55000}{778} = 70 \cdot 69.$$

$$\text{Increase of Entropy} = \frac{70 \cdot 69}{300 + 461} = \cdot 092.$$

Referring to Example 1,

Then, $$\text{Foot-lbs. of Work done per lb. steam} = \frac{\frac{W}{1} \times (B + C) \times S}{32 \cdot 2}.$$

Observe that S = travel of rotor in feet, so that lbs. force × feet travelled = foot-lbs. of work done.

Referring to Example 2,

Then, $$\text{Foot-lbs. of Work done per lb. steam} = \frac{\frac{W}{1} \times (B - C) \times S}{32.2}.$$

Superheating Effect of Blade Friction Losses.

The heat energy lost by way of friction in nozzles or blades reappears later on as "regenerated" heat, and results in drying or re-evaporating the steam at further expansion positions towards the exhaust end of the turbine. This addition of heat units alters the condition of the steam from that of purely "adiabatic" expansion and on the entropy diagram throws out the expansion line as a curve to the right instead of the usual vertical line. This difference in position of the actual expansion line is dependent on the efficiency of the blades or nozzles, as the loss of efficiency represents approximately the gain in steam re-evaporation as described.

The additional heat units recovered in this way may be approximately determined as follows:—

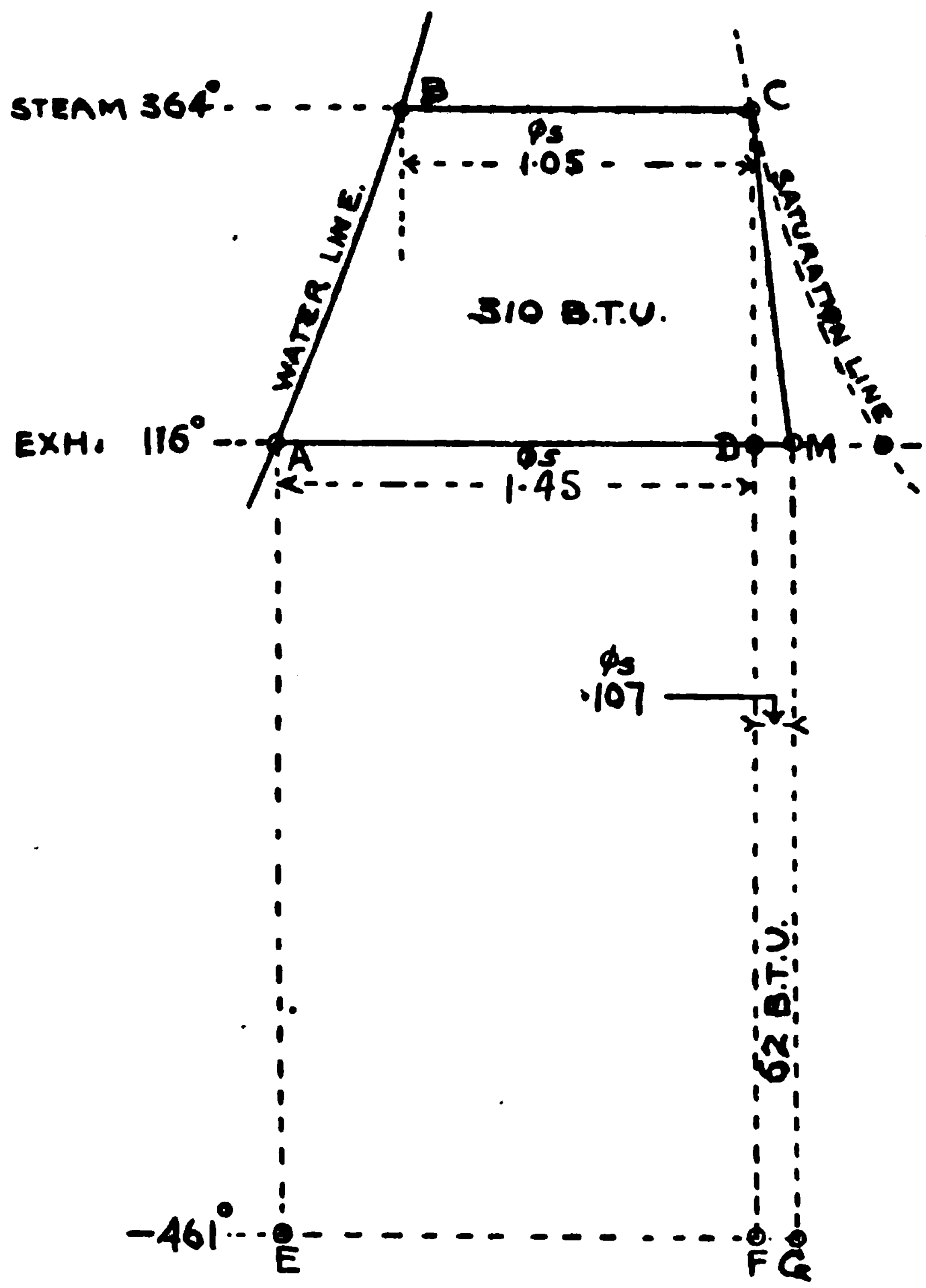
$$\text{Available B.T.U.} \times (1 - \text{Efficiency}) = \text{B.T.U. recovered for re-evaporation.}$$

Example 1.—The available heat units measured on an entropy diagram = 310 B.T.U. (Rankine cycle), and the nozzle efficiency = 80 per cent. (See page 349.)

Find the heat units available for re-evaporation at lower pressure stages of the turbine if the exhaust temperature = 116°.

Then, $$310 \times (1 - \cdot 8) = 310 \times \cdot 2 = 62 \text{ B.T.U. Ans.}$$

Example 2.—From the foregoing determine the position of the actual expansion line M on the entropy diagram, if the initial steam pressure is 145 lbs. gauge, and the temperature 364° (36).



No. .—Entropy Diagram.

Showing Reheating Effects of Friction Losses.

Now, B.T.U. = Temperature Difference × Entropy.

Then, $62 = (116° + 461°) \times \text{Entropy} = 577 \times \text{Entropy}$.

Therefore, $\text{Entropy} = 62 \div 577 = \cdot 107$.

Measure then from the adiabatic line D and to the right ·107 entropy which gives the point of approximate actual turbine expansion

To determine additional B.T.U. obtained by heat regeneration—

Rule, B.T.U. = Temperature Difference × Entropy (mean).

Then, $\text{B.T.U.} = (364° - 116°) \times \left(\frac{\cdot 107 + 0}{2}\right)$.

$$\text{B.T.U.} = 248 \times \cdot 0535 = 13\cdot 26 \text{ B.T.U.}$$

Note.—The small triangular area C.D.M. is o in width at top and ·107 at bottom, so that the mean of these must be taken—

The total heat units available are therefore equal to—

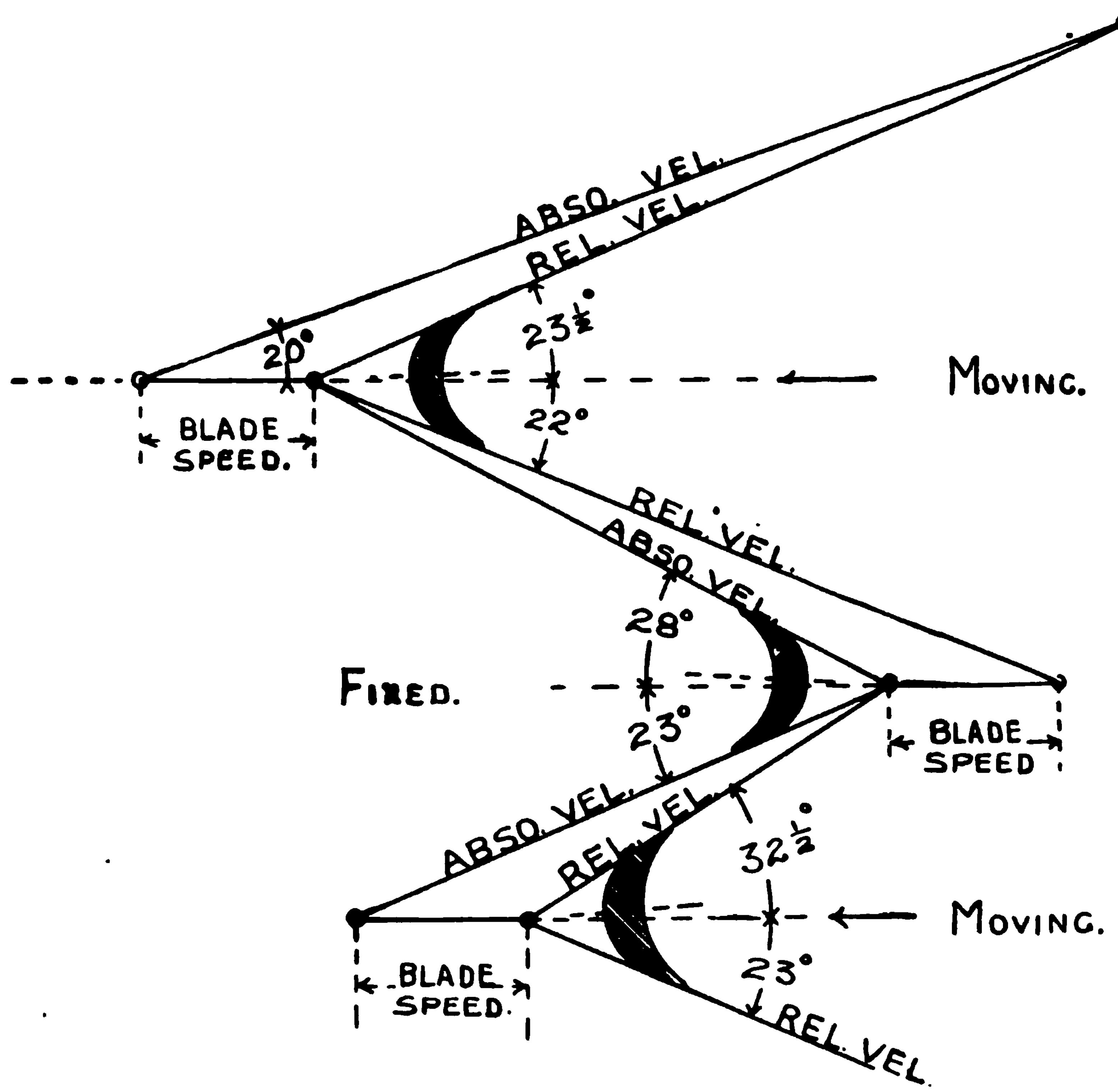
$$310 + 13{\cdot}26 = 323{\cdot}26 \text{ B.T.U.}$$

Blade Angles and Steam Velocity.

To reduce losses by shock :—

(1) The blade angle at admission edge depends on the angle of the relative steam velocity entering the blades.

(2) The blade angle at exit edge depends on the angle of the relative steam velocity leaving the blades.



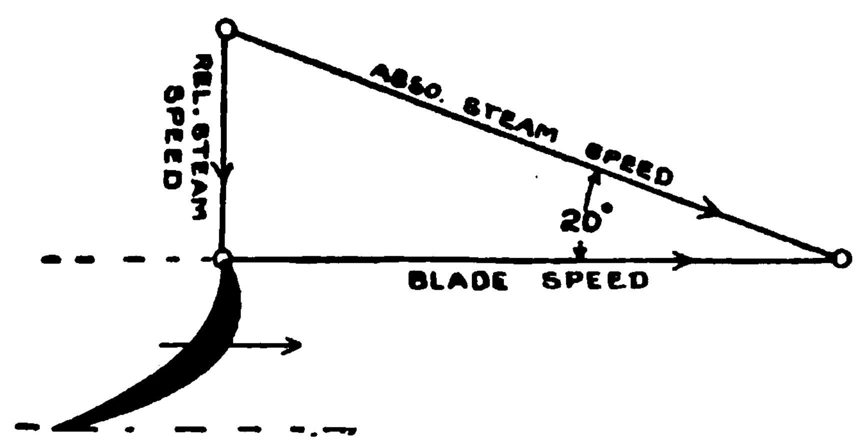
No. 37.—Diagram of Velocities for Impulse Blading.

The blade angles marked are from actual practice.

In practice, however, these conditions require modification, as the necessary thickness of the blade itself interferes with the ideally correct shape of blade curvature, and usually a compromise is arrived at in the actual blade form actualiy decided on.

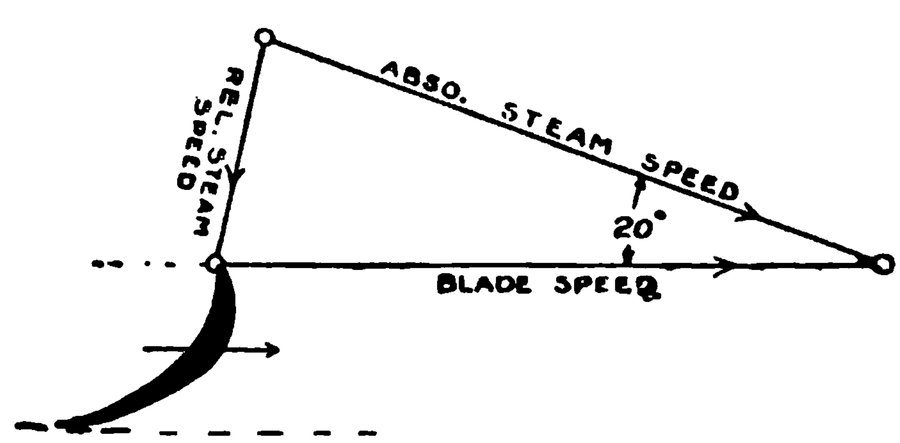
Excessive Blade Speed Compared to Steam Speed.

If the ratio of blade speed to steam speed is excessive the result may be as shown in the sketch (38), where it will be noticed that the steam is entering the blades in streams parallel to the shaft axis, and therefore exerting no impulse effort on the moving blades.



No. 38.

If the blade speed is more excessive still (relatively), then the effect may be as shown (39) and the rotative effect become negative, as the steam would then strike the back of the blades.



No. 39.

Values of Heat Losses.

The average heat losses are made up as follows:—

Steam friction in nozzles - - -	14 per cent.
,, ,, blades - - -	8 ,,
Steam exit velocity losses - - -	10 ,,
Additional general losses - - -	16 ,,
	48 per cent.

Therefore, Efficiency = 100 — 48 = 52 per cent.

Example.—Initial pressure 200 lbs. gauge, dryness 1, pressure at first stage 80 lbs. gauge, dryness ·96. Find steam speed and shaft

revolutions required if there are three rows of moving blades on the wheel, the mean diameter across blades being 7 ft., neglect blade friction and blade shock losses.

Then, $200 + 15 = 215$ lbs. absolute $= 387{\cdot}5^\circ$ and Latent Heat $= 840$ } From
,, $80 + 15 = 95$,, ,, $= 323{\cdot}8^\circ$,, ,, ,, $= 886$ } Tables.

Again, $$387{\cdot}5^\circ + 461^\circ = 848{\cdot}5,$$

And, $$323{\cdot}8^\circ + 461^\circ = 784{\cdot}8.$$

$$\text{Heat Drop} = 840 \times 1 - 886 \times {\cdot}95 + 848{\cdot}5 - 784{\cdot}8 = 62 \text{ B.T.U.}$$

$$\text{Steam Velocity} = \sqrt{62 \times 64{\cdot}4 \times 778} = 1762 \text{ ft. per sec. (nearly),}$$

for single row of blades and half reaction.

Then, $$1762 \div 2 = 881 \text{ ft. blade speed per sec.}$$

As, however, there are three rows of blades in the wheel,

Then, $$\frac{881}{3} = 293 \text{ ft. per sec. blade speed,}$$

Therefore, $$\text{Revolutions} = \frac{293 \times 60}{7 \times 3{\cdot}1416} = 800 \text{ revs. per min.}$$

In actual practice the steam speeds and blade speeds are much less than the above, owing chiefly to friction, blade shock, and other losses, the blade speeds ranging from about 150 to 200 ft. per sec., with steam speeds in proportion.

Available Heat.

By this is meant the total B.T.U. obtained by finding the mean entropy between the working temperature limits of the turbine and which, multiplied by the difference of temperature, gives the B.T.U. available, neglecting all heat losses.

Example.—Find the total B.T.U. available (assuming adiabatic expansion) if the initial steam is 180 lbs. gauge (dryness = 1) and the terminal pressure 1·5 lbs. absolute, and neglecting heat regenerated from friction losses.

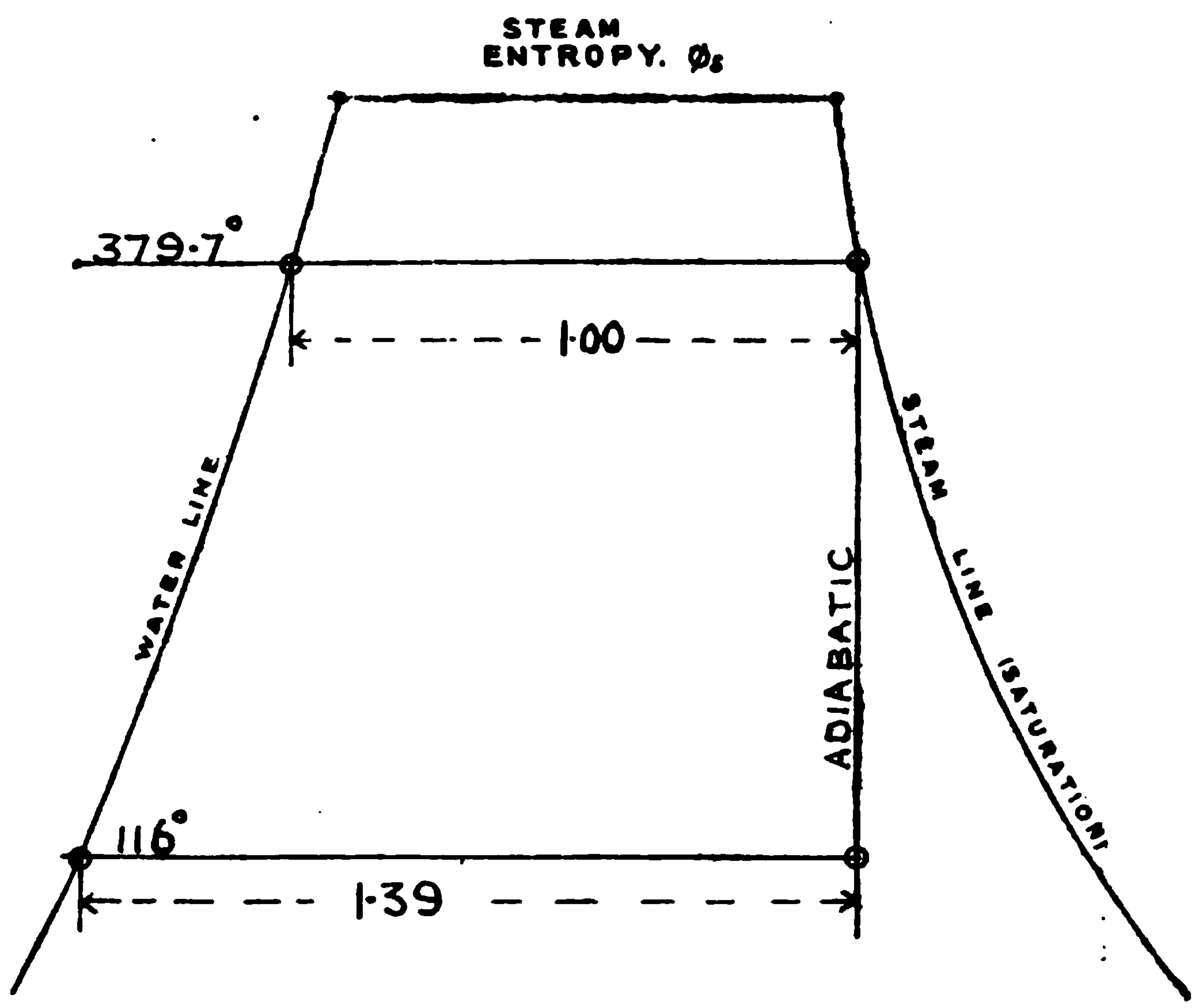
Then, $180 + 15 = 195$ absolute $= 379{\cdot}7^\circ$ Temperature.

And, $1{\cdot}5$,, $= 116^\circ$,,

On drawing lines across the entropy diagram (40) at these temperature levels, the entropy will be found to be 1 at the higher temperature and 1·39 at the lower temperature.

Therefore, $$\text{B.T.U. available} = \left(\frac{1 + 1{\cdot}39}{2}\right) \times (379{\cdot}7 - 116) = 314{\cdot}92 \text{ B.T.U.}$$

As explained on page 391, the actual B.T.U. available is slightly in excess of this, owing to the increase of entropy at the lower temperature level due to regenerated heat produced by blade frictional losses.



No. 40.—Entropy Diagram.

Pressure Drop in Stages.

As the latent heat value of steam increases with fall of pressure, the pressure drop to give a required heat drop per stage is much more at the first stage than at any of the following stages; this can be tested satisfactorily by means of the entropy diagram.

In the first stage the pressure may require to fall from 180 lbs. to 70 lbs. to give the heat drop necessary to obtain the required nozzle steam velocity, whereas to obtain equal steam velocity in the second stage nozzles a pressure drop of from 70 lbs. to 40 lbs. or less may be sufficient to obtain equal steam velocity, and similarly, for the other stages. The pressure drop required is also influenced directly by the dryness condition of the steam at the various stages.

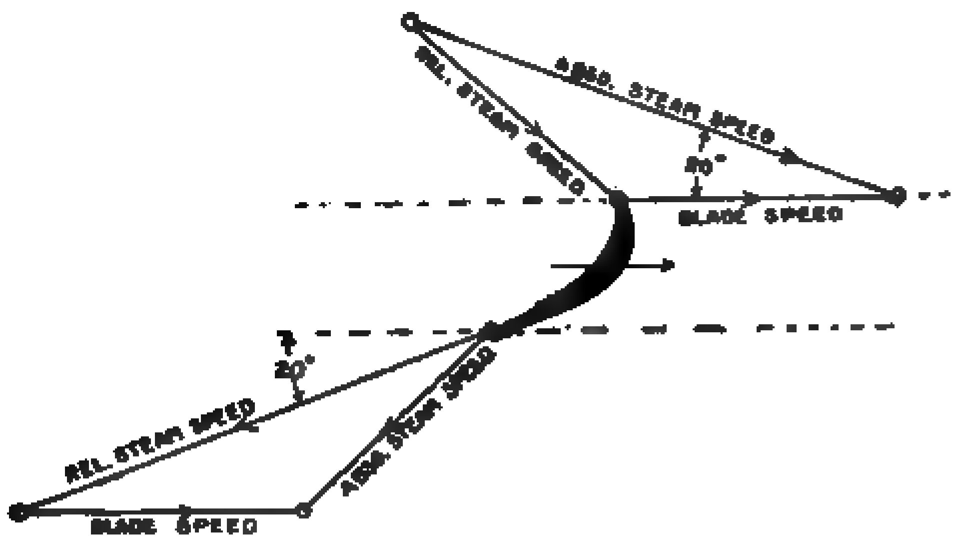
The following points should be noted.

Impulse Stage.

For any stage the pressure is approximately equal at initial end and exit end, therefore blade tip steam leakage is practically eliminated. The guide of fixed blades merely deflect the flow of steam without changing its pressure or velocity to any appreciable extent.

No. 41.—Combined Impulse and Reaction Turbine.

The impulse wheel is supplied with two moving blade rows and one fixed row.



No. 42.—Velocity Triangles of Reaction Blading.

steam, also cause a fall in pressure and an increase in
accompanied by a corresponding heat drop.

PRESSURE
LINE

VELOCITY
CURVE

No. 43.—Pressure and Velocity Curves.

Impulse Blading.

Notice that the pressure remains constant in both moving and fixed blades, but the velocity falls in the moving blades and remains constant in the fixed blades.

Reaction Stage.

Pressure less at exit end, therefore blade tip clearance leakage takes place; the guide blades, in addition to deflecting the flow of steam, also effect a slight increase in its velocity.

In the moving blades velocity is extracted in the doing of useful work, and velocity is also generated at the same time, accompanied by a fall of pressure and a drop in heat.

PRESSURE
LINE —

CURVE

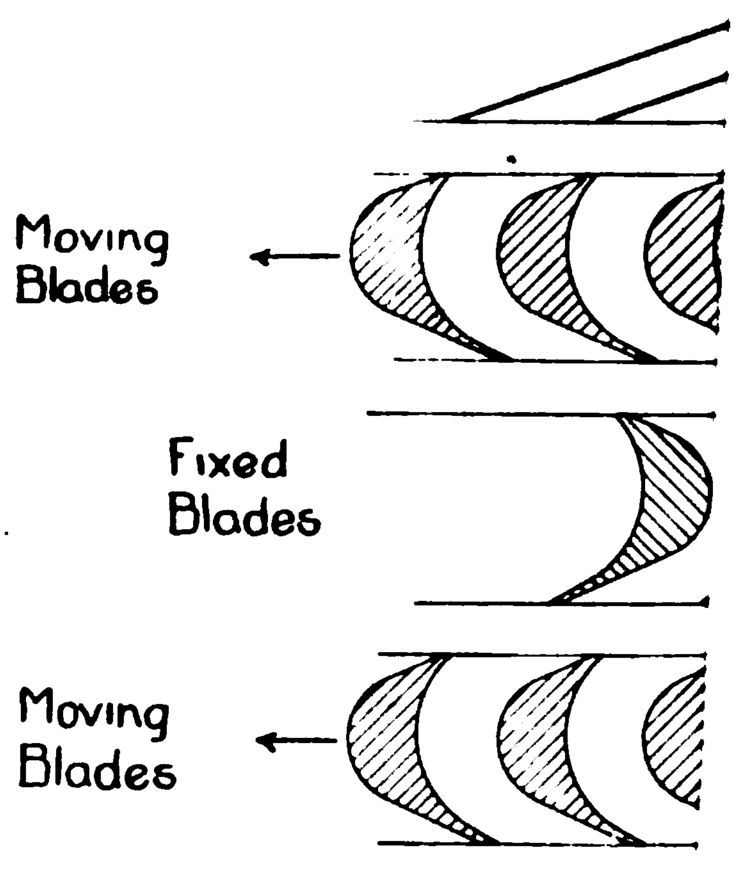
No. 44.—Pressure and Velocity Curves.

Reaction Blading.

Line E shows the fall in velocity which would take place if the blades were of impulse type, but as the exit openings are restricted in reaction blading velocity is also generated, causing a rise in the curve as shown.

Impulse Turbines.

Nozzles -	Fall of pressure „ temperature Heat drop Increase of velocity	Work done on steam to increase its velocity.



(Pressure difference at admission and exit edges.)

blade shock losses, which combined vary from about 20 per cent. to 25 per cent., and depend on the angle and shape of the blades employed.

Impulse Turbines.

Nozzles -	Fall of pressure „ temperature Heat drop Increase of velocity	Work done on steam to increase its velocity.

Moving Blades	Constant pressure Decrease in velocity	Work done on shaft to cause rotation of same.
Guide Blades	Constant pressure „ velocity	No work done. Path of steam merely redirected.

Reaction Turbines.

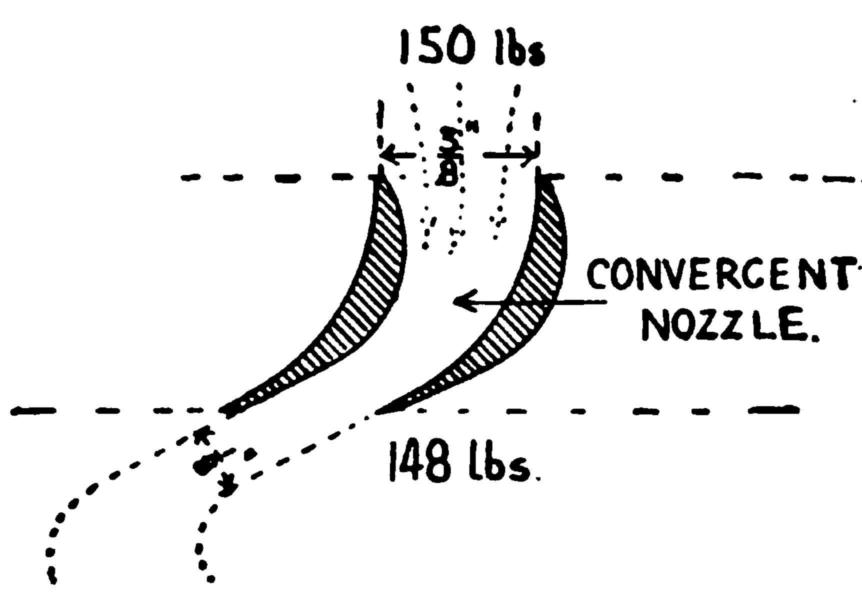
Guide Blades	Fall of pressure „ temperature Heat drop Increase of velocity	Work done on steam to increase its velocity.
Moving Blades	Fall of pressure. „ temperature. Heat drop Decrease of velocity due to work done on shaft. Also, increase of velocity due to the nozzle-like effect of blade openings.	

Degree of Reaction.

By this is meant the amount of energy or velocity still contained in the steam after it leaves the guide passages. So that if, say, the "degree of reaction" = ·5, the blade speed is then equal to half the speed of the steam.

Blade Shock and Friction Losses.

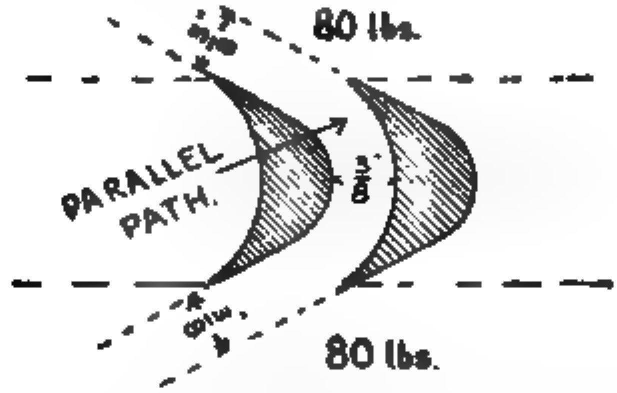
The efficiency of all turbines is reduced by blade friction and



No. 46.—Sketch of Reaction Blading showing Convergent Nozzle Effect of Blades.

(Pressure difference at admission and exit edges.)

blade shock losses, which combined vary from about 20 per cent. to 25 per cent., and depend on the angle and shape of the blades employed.



No. 47.—Sketch of Impulse Blading showing Parallel Path of Steam Flow.

(No pressure difference at admission and exit edges.)

Taper of Nozzles.

The taper varies in different types from 1 in 12, to 1 in 26.

In marine turbines the nozzle passages are rectangular in section, the depth of opening corresponding more or less with the height of the first moving blade rows.

The nozzle openings are arranged in the arc of a circle on the upper half circumference of the turbine casing, and are divided off

3. Impulse blading.
4. First expansion reaction blading.
5. Second expansion reaction blading.
6. Third expansion reaction blading.
7. Auxiliary exhaust admission to third expansion for "closed exhaust" working.

in number by division plates, the steam flow to each set so divided being controlled by hand-operated valves. This arrangement is known as "partial admission."

No. 50—Sketch showing Nozzle Openings and Impulse Guide Blade Positions at 1st, 2nd, or 3rd Stage Expansions of Curtis Turbines.

Note.—**In Parson's impulse blading the guide blades often extend round the complete half circumference of the top casing.**

Reasons for Increase of Blade Heights.

Reaction Blading.—As the pressure falls continuously throughout the turbine, and the steam volume increases in like proportion, the blade heights have, therefore, to be increased to allow of equal weight of steam flow in a given time, the usual blade increase ratio being as $1 : \sqrt{2}$ or as $1 : 1{\cdot}42$.

Impulse Blading.—In any single stage of an impulse turbine the pressure and therefore the steam volume remains almost constant throughout that stage, but, as the velocity of the steam is reduced at each moving blade row, the blade height must be increased to allow of the same volume of slower moving steam to pass in a given time; this accounts for the regular increase of height of blade in each stage, the increase is, in most cases, nearly the same per blade, as the loss of velocity is almost the same for each blade row; the following example from a Curtis turbine will make this point clear.

Example.—Brown-Curtis turbine, S.H.P. 7000, revs., 550 per min.

1st Stage Wheel of Four Blade Rows.

1st Row, blade height = $1\frac{1}{16}$ in.	Difference in height in each row = $\frac{14}{32}'' = \frac{7}{16}''$.
2nd " " " = $1\frac{1}{2}$ "	
3rd " " " = $1\frac{15}{16}$ "	
4th " " " = $2\frac{3}{8}$ "	

Note.—Diameter of blade pitch circle at 1st row = 76 in.

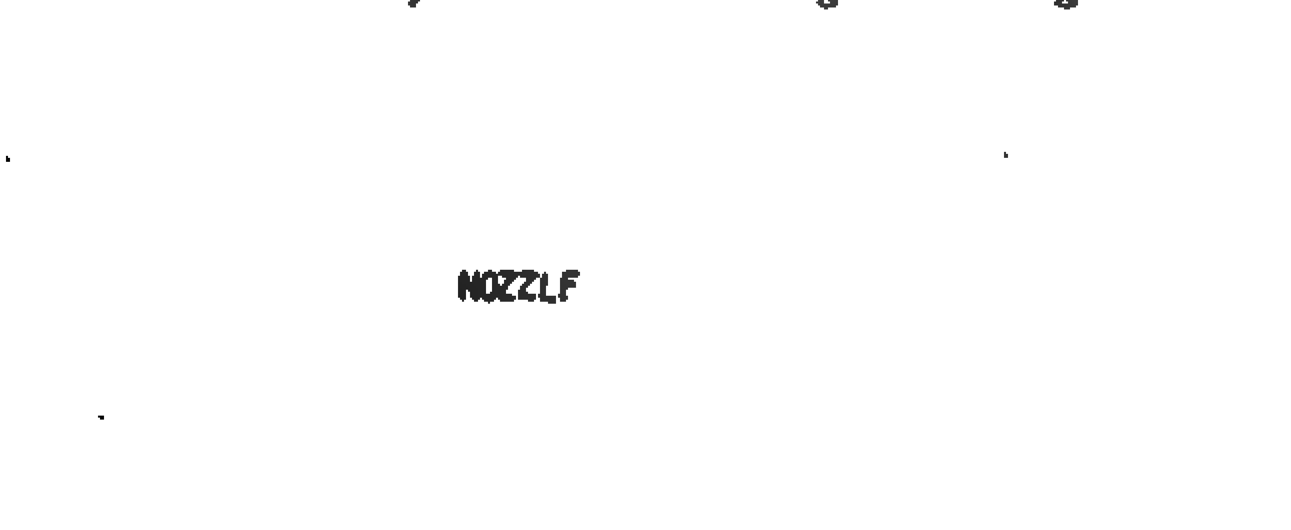
5th Stage Wheel of Three Blade Rows.

1st Row, blade height = $1\frac{5}{16}$ in.	Difference in blade height in each row = $\frac{6}{16}'' = \frac{3}{8}''$.
2nd " " " = $1\frac{11}{16}$ "	
3rd " " " = $2\frac{1}{16}$ "	

Note.—Diameter of blade pitch circle at 1st row = 73 in.

Difference in Velocity and Blade Height.

A reference to the diagram of blade velocities (p. 392) shows that the absolute velocity of the steam leaving the moving blades is less



No. 51.—Blade Stepping.

than the relative velocity of the steam entering the blade, and to allow for this the blade heights should be more at the exit edges.

In the Curtis and other impulse turbines this difference is usually provided for by the combined stepping on a taper of the blades, the wheel, and the casing, as shown in the sketch (51).

Moving Blades.

The admission angle is dependant on the *relative* steam admission velocity.

The exit angle is dependant on the *relative* steam exit velocity.

Fixed Blades.

The admission angle is dependant on the *absolute* steam admission velocity.

The exit angle is dependant on the *absolute* steam exit velocity.

As stated elsewhere, to obtain maximum blade efficiency the admission angle is usually made more than the exit angle.

Sketch No. (52) shows the usual blading arrangement of an impulse turbine, and the diagram (37) shows clearly the absolute and relative velocities referred to, compared with the blade angles at admission and exit edges.

Blade Admission and Exit Angles.

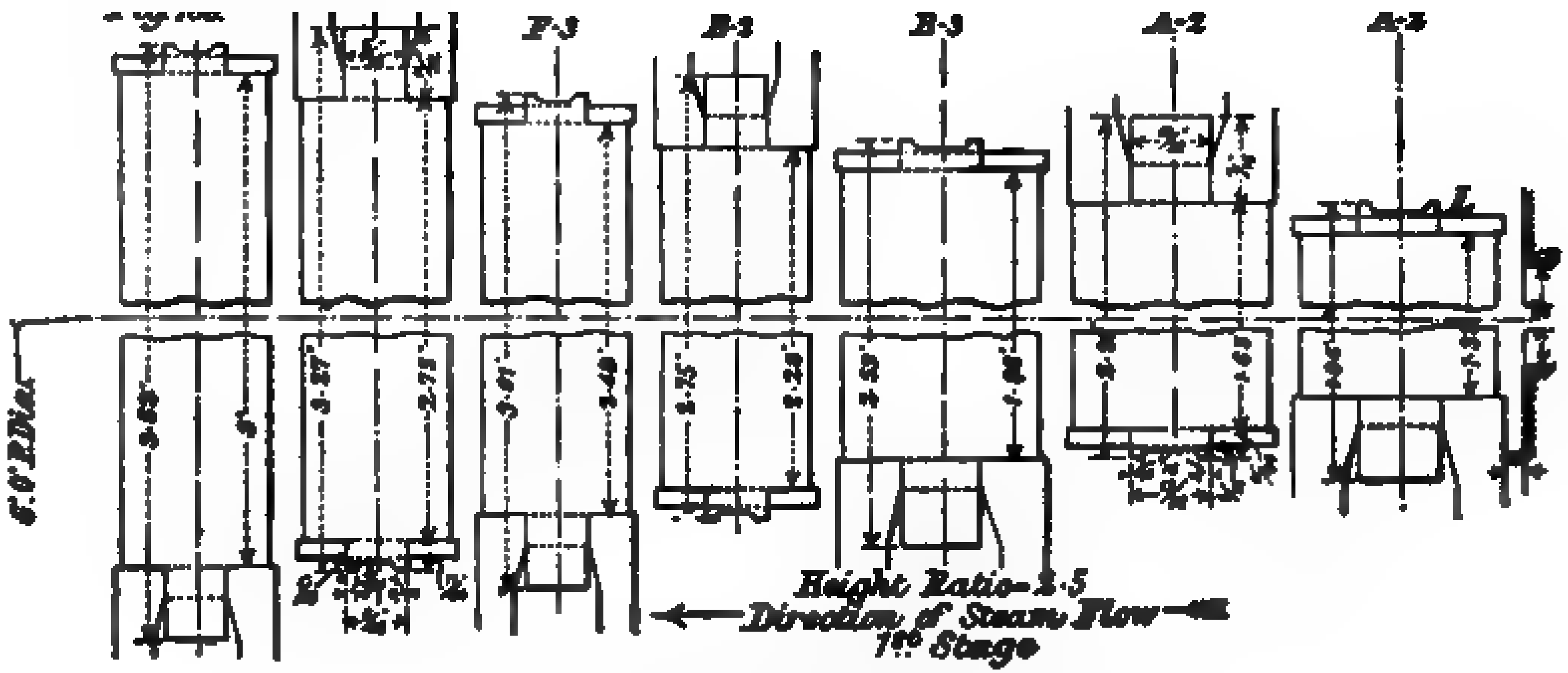
In Curtis turbine practice the blade exit angle (52) is made less than the blade admission angle, this being arranged for by placing the blades out of line axially; that is, a slight cant is given to the blade position, which has the effect of increasing the angle at steam admission and proportionally decreasing the angle of exit. This arrangement tends to increase the loss by "blade shock", but evidently the advantages obtained in other ways more than make up this loss: were it otherwise this practice would not be adhered to.

Blade Angles—1st Stage.

The following table gives the angles of the blades at entry and exit edges for the 1st stage of seven blade rows of a Curtis turbine.

	Blade Angle at Entrance.	Blade Angle at Exit.
1st Blade Row (moving) - -	20°	22°
2nd " " (fixed) - - -	28°	22°
3rd " " (moving) - -	34°	22°
4th " " (fixed) - - -	34°	24°
5th " " (moving) - -	42°	23°
6th " " (fixed) - - -	42°	23°
7th " " (moving) - -	68°	45°

Referring to the illustration (52) it will be noticed that the blades do not lie square to the line of the turbine, the exit edge overlapping in every case; hence the difference in the entrance and exit angles. This reduction in the angle of exit requires longer blades, as the less the angle, the less the area open for steam flow.

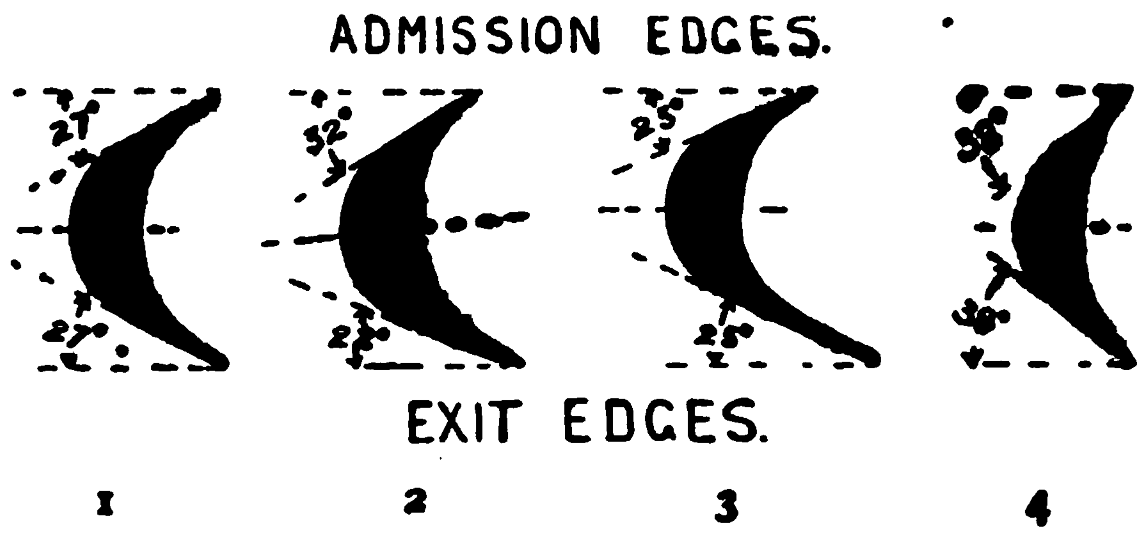


No. 52.—1st Stage Impulse Blading.

(4 Moving Rows, 3 Fixed Rows.)

(U.S. Destroyers, "Perkins" and "Sterrett.")

Observe that the blades are all slightly canted to give a greater angle at admission than at exit.



No. 53.—Impulse Blade Angles.

1. Blade with equal angles at admission and exit.
2. ,, canted in position and giving unequal angles at admission and exit.
3. ,, ,, "tail" continuation.
4. ,, reduced in thickness and giving greater angles of admission and exit.

From the foregoing it will be obvious that the exit edge angle determines the angle of steam velocity, also that the admission edge angle may be made to agree with this, or may be made not to agree with it as practical considerations require.

Area of Nozzle Openings.

$$\text{Throat area} = \frac{\text{Lbs. flow per sec.} \times \text{Actual Volume} \times 144 \text{ sq. in.}}{\text{Velocity at throat}} = \text{Inches area.}$$

$$\text{Mouth area} = \frac{\text{Lbs. flow per sec.} \times \text{Actual Volume} \times 144 \text{ sq. in.}}{\text{Velocity at mouth}} = \text{Inches area.}$$

$$\text{Clear total Length at mouth} = \frac{\text{Inches area}}{\text{1st stage blade Heights}} = \text{Inches.}$$

$$\text{Length of each nozzle} = \frac{\text{Inches length}}{\text{Number of nozzles}} = \text{Inches.}$$

Note.—The division walls between each nozzle vary in thickness from ·06 in. to ·1 in.

Actual Volume.

Specific Volume × Dryness Constant = Actual Volume.

The dryness value for any pressure drop can be obtained from the entropy diagram, as shown clearly in the section dealing with this subject (p. 348).

Example.—

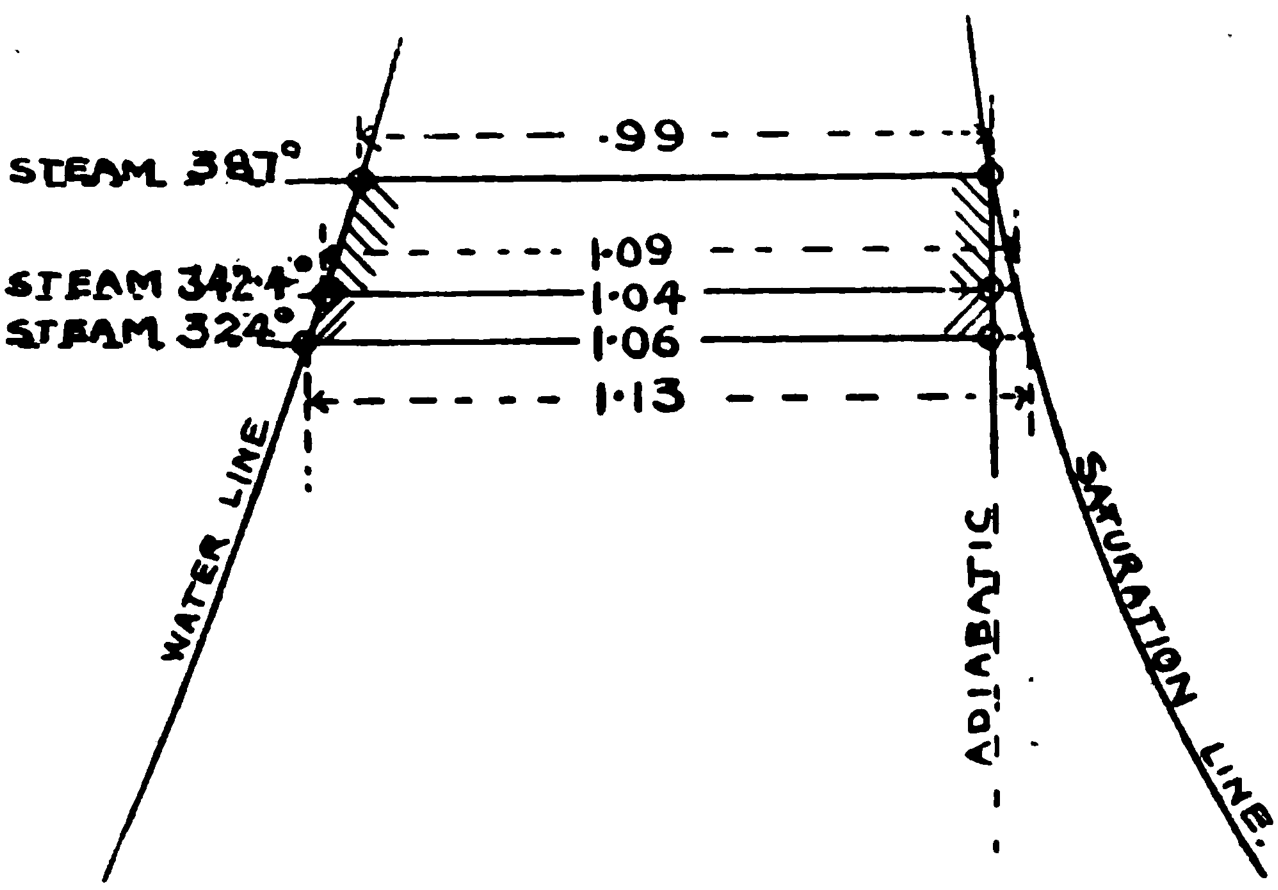
Steam flow per sec. = 50 lbs.	Find required dimensions of nozzles.
Initial Steam Pressure = 195 lbs. gauge.	
Nozzle Efficiency = ·9.	
Condition of Steam . . Dry.	

Then, 195 + 15 = 210 lbs. absolute and Temperature 387°.

Throat Pressure = 210 × ·58 = 121·8 lbs. absolute, say 122 lbs.

122 lbs. absolute = 3·59 cub. ft. specific volume and temperature 342·4°.

Assuming adiabatic expansion, on the entropy diagram (54) draw horizontal lines at the two temperature levels of 387° and 343·5°, drop a vertical from the higher to the lower, and find the mean entropy; also find the dryness as shown.



No. 54.—Entropy Diagram.

At 387° Temperature Entropy = ·99

„ 342·4° „ „ = 1·04

$$\text{Mean Entropy} = \frac{·99 + 1·04}{2} = 1·015.$$

Temperature Difference = 387° − 342·9° = 44·6.

So that, B.T.U. = 1·015 × 44·6 = 45·26.

$$\text{Steam Velocity at Throat} = \sqrt{45·26 \times 64·4 \times 778 \times ·9} = 1421 \text{ ft. per sec.}$$

$$\text{Dryness} = \frac{1·04}{1·09} = ·954.$$

Actual Steam Volume = 3·59 × ·954 = 3·42 cub. ft.

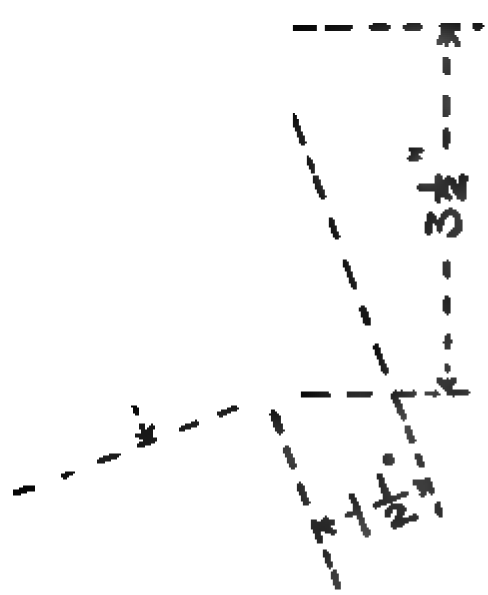
$$\text{Then, Throat Area of Nozzles} = \frac{50 \times 3·42 \times 144}{1421} = 17·3 \text{ sq. in.}$$

Pressure at mouth = 80 lbs. gauge.

Then,

80 + 15 = 95 lbs. absolute, Temperature 324°, specific volume 4·55 cub. ft. (from Table).

As before, set off a line across the entropy diagram at the level of 324° and measure the entropy for adiabatic expansion, which = 1·06.



No. 55.—Impulse Nozzle.

$$\text{Mean Entropy} = \frac{\cdot 99 + 1\cdot 06}{2} = 1\cdot 025.$$

$$\text{Temperature Difference} = 387^\circ - 324^\circ = 63.$$

$$\text{Dryness} = 1\cdot 06 \div 1\cdot 13 = \cdot 938.$$

Then,

$$\text{B.T.U.} = 1\cdot 025 \times 63 = 64\cdot 57$$

So that,

$$\text{Velocity at Mouth (theoretical)} = \sqrt{64\cdot 57 \times 64\cdot 4 \times 778 \times \cdot 9} = 1706 \text{ ft. per sec.}$$

$$\text{Actual Steam Volume} = 4\cdot 55 \times \cdot 938 = 4\cdot 26 \text{ cub. ft.}$$

$$\text{Mouth area} = \frac{50 \times 4\cdot 26 \times 144}{1706} = 17\cdot 9 \text{ sq. in.}$$

Assume that 12 nozzles are to be fitted, the depth of each being, say, 1·25 in.

$$\text{Then, Area of each nozzle at mouth} = \frac{17 \cdot 9}{12} = 1 \cdot 5 \text{ sq in. (nearly).}$$

$$\text{Width of nozzles} = \frac{1 \cdot 5}{1 \cdot 25} = 1 \cdot 2 \text{ in. say } 1\tfrac{1}{4} \text{ in.}$$

The extension of mouth required to bring the nozzles square with the blade rows is made parallel, and the correct size can be found by scale as shown above, and which is equal to $3\frac{1}{2}$ in.

Note.—In practice a few additional nozzles over the number calculated are usually fitted to allow for extra power.

Impulse Blade Heights.

Rule.—

$$W \times \text{Volume of steam} \times 144 = H \times N \times L \times V_3 \times \sin a \times \cdot 9.$$

Where, W = Weight of steam flow per sec.

144 = Square inches per square foot.

H = Height of Blades at exit edge.

N = Number of nozzles.

L = Length of nozzles at mouth (total).

V_3 = Exit velocity of steam.

a = Angle of Blades.

·9 = Constant to allow for loss of area through blade thickness.

Example.—Find required height of first stage impulse blading if the steam flow is 50 lbs. per sec., the steam volume 5 cub. ft. per lb., and the exit velocity 1500 ft. 16 nozzles are fitted each $3\frac{1}{2}$ in. wide at mouth, the blade angle being 22°.

$$\text{Then,} \quad H = \frac{50 \times 5 \times 144}{16 \times 3 \cdot 5 \times 1200 \times \cdot 3746 \times \cdot 9} = 1 \cdot 58 \text{ in., say, } 1\tfrac{19}{32} \text{ in.}$$

Note.—Refer to "trigonometrical table," p. 542, where the sine of angle 22° will be found to be ·3746. For the other blade rows of the same stage the same method holds good, the only change in the formula being the velocity of the steam (which will be less row after row) and the blade angles, the sine of which will increase. As, however, the fall in velocity is more in proportion than the increase of opening due to blade angle increase, the blade heights will work out higher for each row successively.

Nozzle Calculations.

The following calculations show how the dimensions of the nozzles are determined.

Data.—

Pressure at 1st stage, 80 lbs. gauge. Dryness, ·93.
Velocity at end of taper = 1600 ft. per sec.
S.H.P., 10000.
Steam consumption = 14 lbs. per S.H.P. per hour.
Angle of nozzle plates, 20°.
Depth of nozzle openings to be $\frac{7}{8}$ in.
30 nozzles to be fitted.
Then, to find width of nozzle openings at mouth.

Then, $80 + 15 = 95$ lbs. absolute $= 4{\cdot}55$ cub. ft. specific volume (from Table).

$$\text{Actual Volume} = 4{\cdot}55 \times {\cdot}93 = 4{\cdot}23 \text{ cub. ft.}$$

$$\text{Steam flow per second in cubic feet} = \frac{10000 \times 14 \times 4{\cdot}23}{60 \times 60} = 164{\cdot}5 \text{ cub. ft.}$$

$$\text{Combined area of all nozzles} = \frac{164{\cdot}5 \times 144}{1600} = 14{\cdot}8 \text{ sq. in.}$$

$$\text{Area of one nozzle} = \frac{14{\cdot}8}{30} = {\cdot}49 \text{ sq. in.}$$

$$\text{Width of one nozzle at end of taper} = {\cdot}49 \div {\cdot}875 = {\cdot}56 \text{ in.}$$

$$\text{Width of each nozzle at mouth} = {\cdot}56 \div {\cdot}372 = 1{\cdot}5 \text{ in.}$$
(Pitch of nozzle plates)

Note.—·372 = sine of angle 20° (from Table, p. 542).

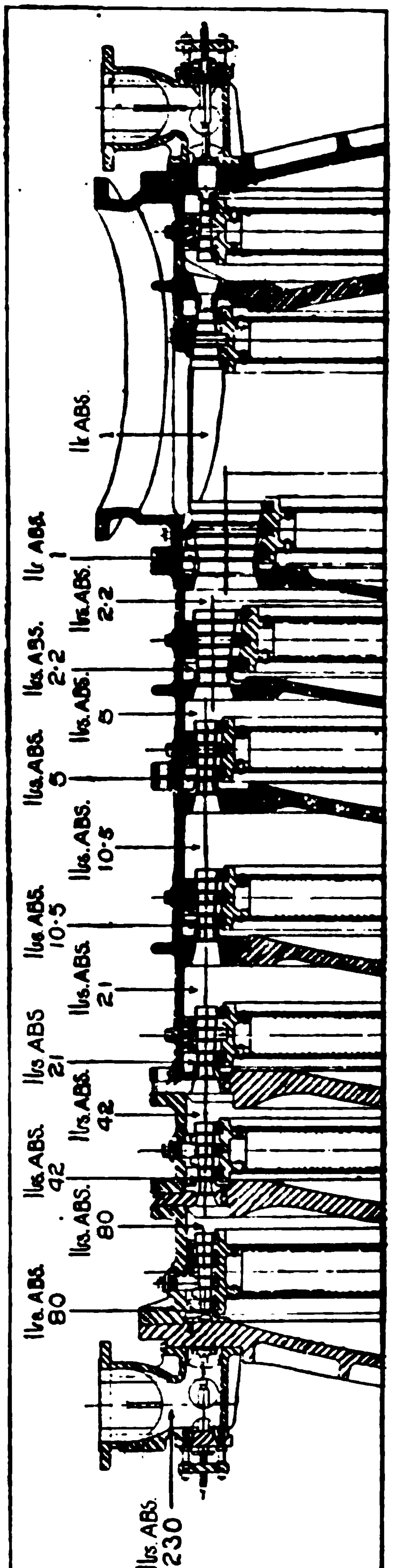
Steam Velocity at Nozzle Stages.

The following example shows approximately how the steam velocity is obtained by means of the heat drop at each successive stage of an impulse turbine of the Curtis type.

Data. 7 ahead stages. Steam initially dry. Nozzle efficiency 85 per cent. (·85) assumed.

PRESSURES IN STAGES.

Stage.	Initial Steam at Nozzle.	Stage Pressure.
1	230 lbs. absolute	80 lbs. absolute
2	80 "	42 "
3	42 "	21 "
4	21 "	10.5 "
5	10.5 "	5 "
6	5 "	2.2 "
7	2.2 "	1 "



No. 56.—Stage Pressures in Curtis Turbine.

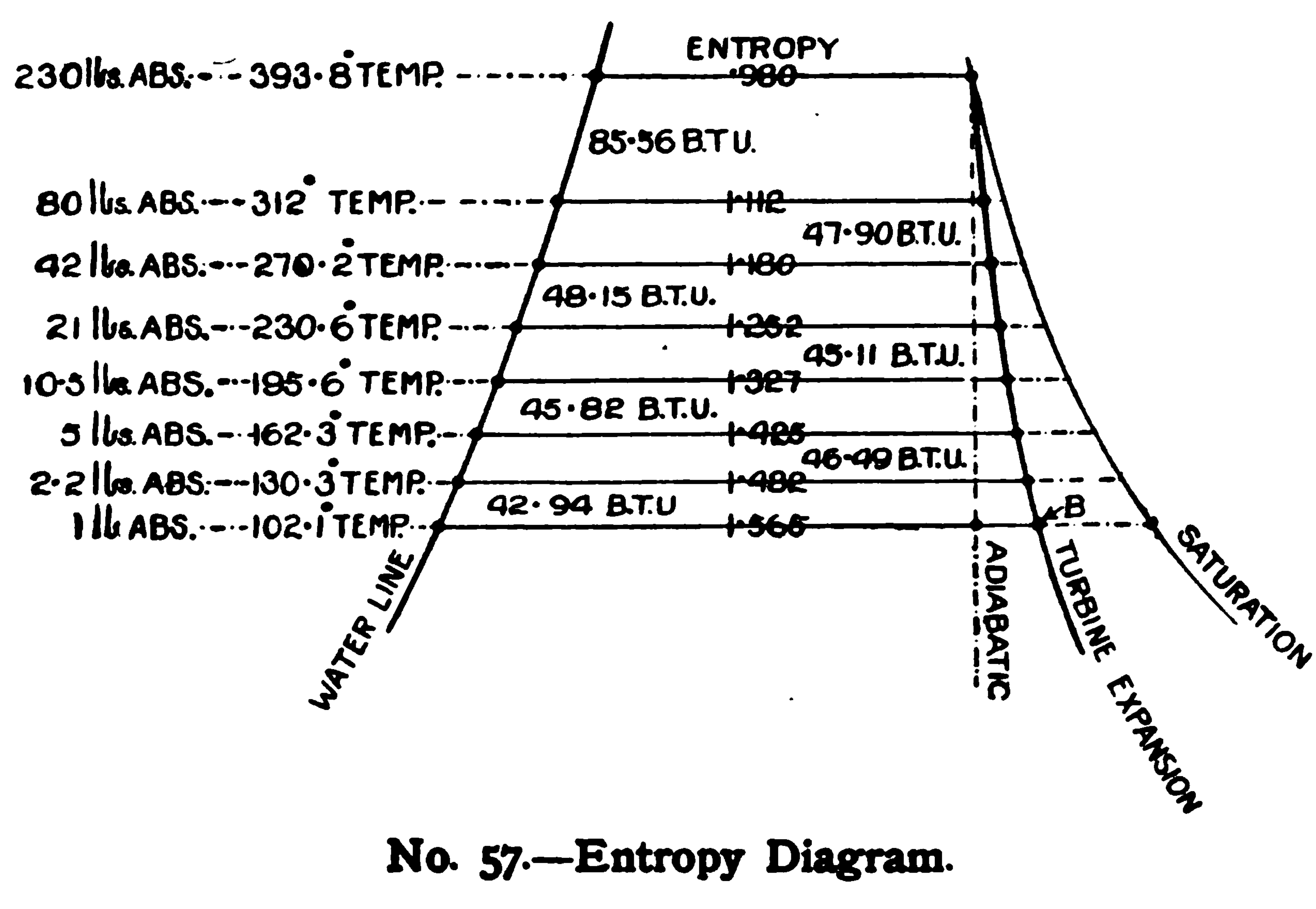
7 Ahead Stages.

Notice that the pressure is constant throughout the blading of each stage, fall of pressure only taking place in the stage expansion nozzles.

Entropy.

Set off lines on the entropy diagram (57) at the temperature levels corresponding to the pressures given, and which are as follows (from Steam Table):—

Absolute Pressure.	Temperature ° Fahrenheit.
230	393.8
80	312
42	270.2
21	230.6
10.5	195.6
5	162.3
2.2	130.3
1	102.1



No. 57.—Entropy Diagram.

Next draw in the turbine expansion line B, which, in this case, has been assumed as occupying a position ·33 of the distance between the adiabatic line and the saturation line.

Note.—The position of this line is due to the reheating effects of blade friction as previously explained (p. 391).

Now measure the entropy values at each temperature level and which read as follows:—

Temperature ° Fahrenheit.	Entropy.
393.8	.980
312	1.112
270.2	1.180
230.6	1.252
195.6	1.327
162.3	1.425
130.3	1.482
102.1	1.565

1ST STAGE.—

B.T.U. drop = Mean Entropy × Temperature Difference.

Therefore, $$\text{B.T.U.} = \left(\frac{\cdot980 + 1\cdot112}{2}\right) \times (393\cdot8 - 312) = 85\cdot56.$$

Steam Velocity at nozzle $= \sqrt{85\cdot56 \times 64\cdot4 \times 778 \times \cdot85} = 1908$ ft. per sec.

2ND STAGE.—

$$\text{B.T.U.} = \left(\frac{1\cdot112 + 1\cdot180}{2}\right) \times (312 - 270\cdot2) = 47\cdot90.$$

Steam Velocity at nozzle $= \sqrt{47\cdot90 \times 64\cdot4 \times 778 \times \cdot85} = 1428$ ft. per sec.

3RD STAGE.—

$$\text{B.T.U.} = \left(\frac{1\cdot180 + 1\cdot252}{2}\right) \times (270\cdot2 - 230\cdot6) = 48\cdot15.$$

Steam Velocity at nozzle $= \sqrt{48\cdot15 \times 64\cdot4 \times 778 \times \cdot85} = 1431$ ft. per sec.

4TH STAGE.—

$$\text{B.T.U.} = \left(\frac{1\cdot252 + 1\cdot327}{2}\right) \times (230\cdot6 - 195\cdot6) = 45\cdot11.$$

Steam Velocity at nozzle $= \sqrt{45\cdot11 \times 64\cdot4 \times 778 \times \cdot85} = 1386$ ft. per sec.

5TH STAGE.—

$$\text{B.T.U.} = \left(\frac{1\cdot327 + 1\cdot425}{2}\right) \times (195\cdot6 - 162\cdot3) = 45\cdot82.$$

Steam Velocity at nozzle $= \sqrt{45\cdot82 \times 64\cdot4 \times 778 \times \cdot85} = 1396$ ft. per sec.

6TH STAGE.—

$$\text{B.T.U.} = \left(\frac{1\cdot425 + 1\cdot482}{2}\right) \times (162\cdot3 - 130\cdot3) = 46\cdot49.$$

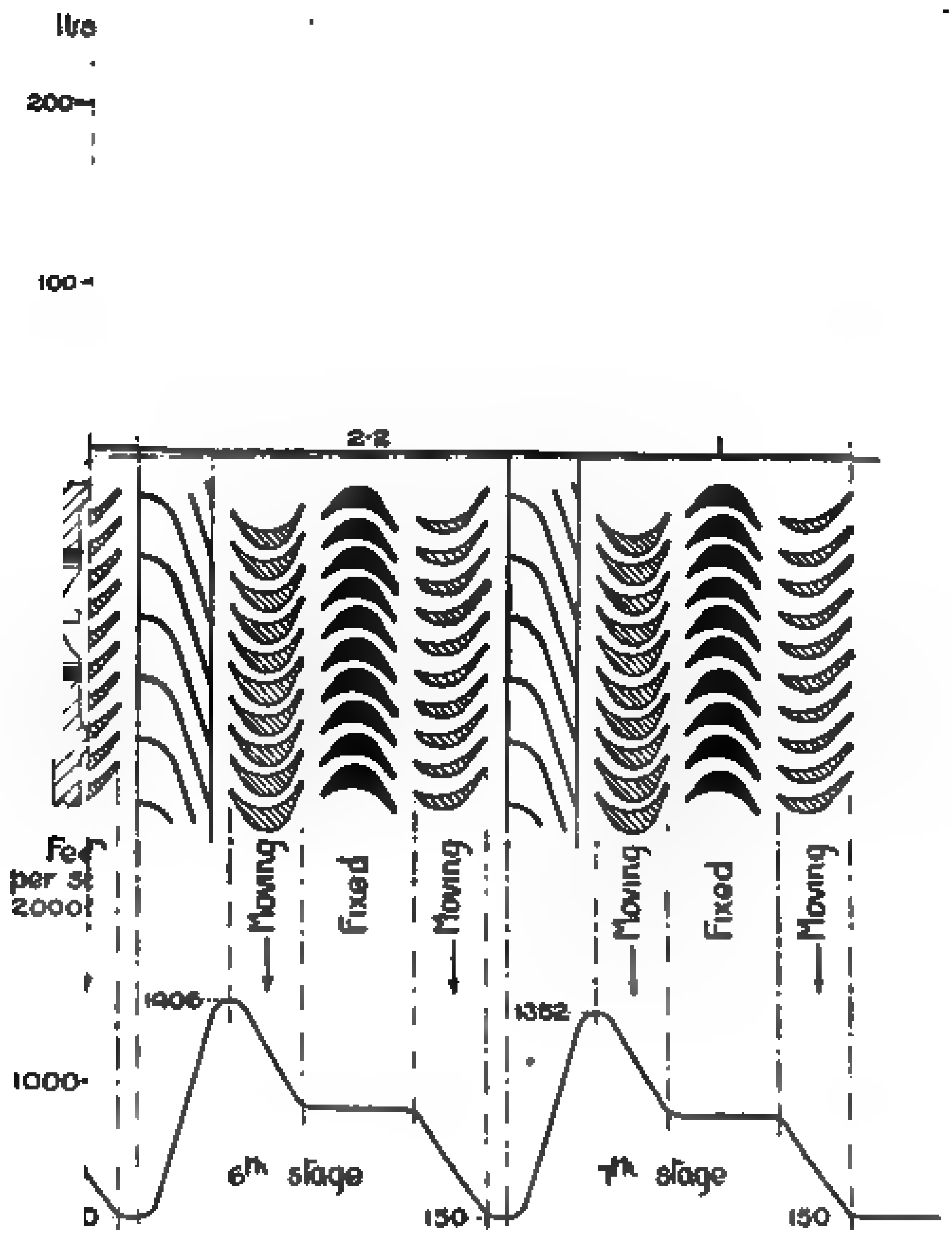
Steam Velocity at nozzle $= \sqrt{46\cdot49 \times 64\cdot4 \times 778 \times \cdot85} = 1406$ ft. per sec.

7TH STAGE.—

$$\text{B.T.U.} = \left(\frac{1\cdot482 + 1\cdot565}{2}\right) \times (130\cdot3 - 102\cdot1) = 42\cdot94.$$

Steam Velocity at nozzle $= \sqrt{42\cdot94 \times 64\cdot4 \times 778 \times \cdot85} = 1352$ ft. per sec.

It should be noted that to give practically equal steam velocity at each nozzle, the pressure drop required is more at the higher



Turbine.

To face page 412.

pressure end than at the lower pressure end of the turbine. An examination of the entropy diagram will make clear the reason for this.

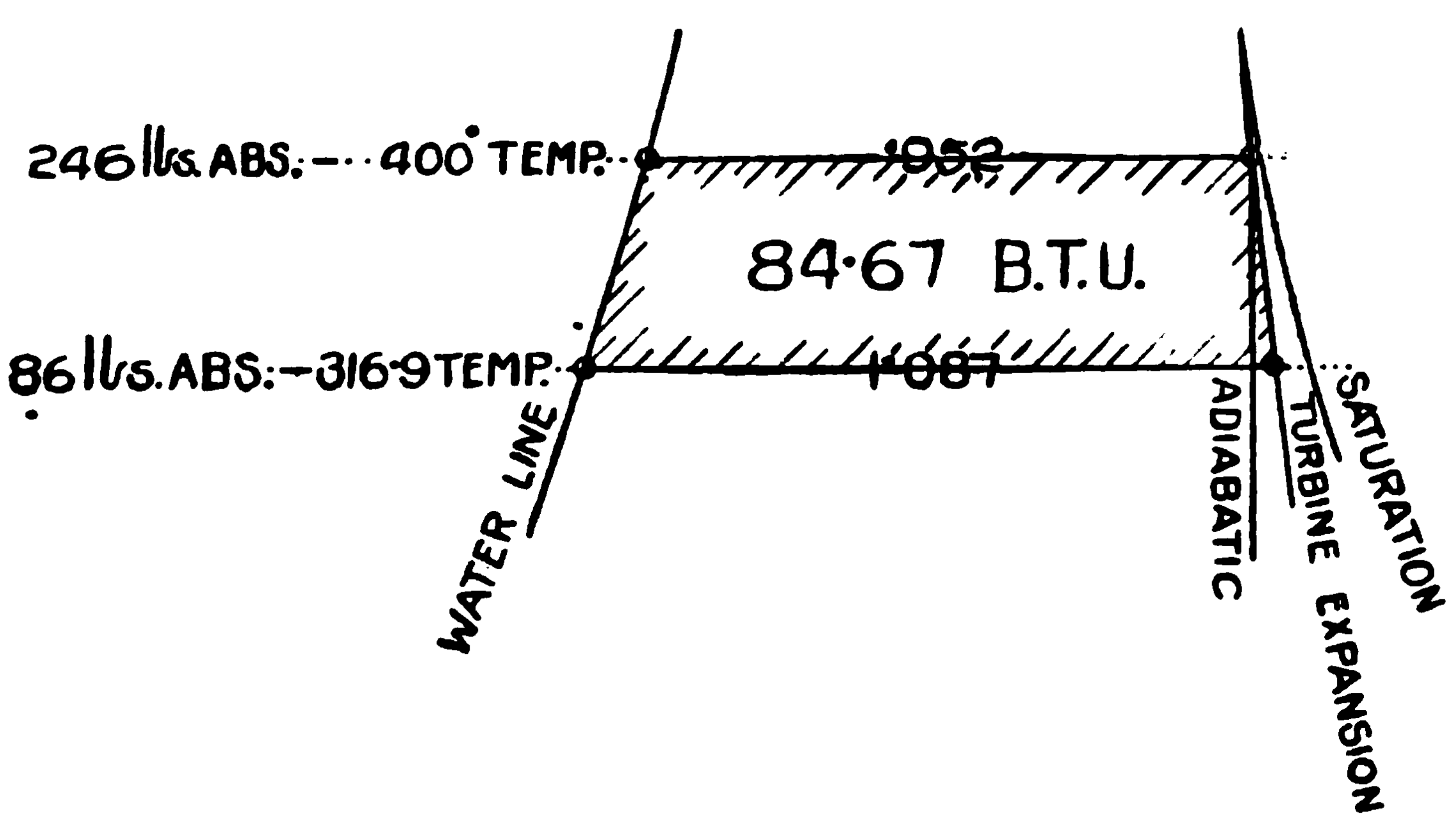
Velocity Diagram for Curtis Turbine (1st Stage).

By way of an example from practice for the student, the following calculations are given for the 1st stage of the Curtis turbines of the U.S. destroyer "Sterrett," the results of which form an instructive comparison between theoretical calculation and practical design.

Data.—

Initial Steam, 246 lbs. absolute = 400° Temperature.
1st stage Pressure, 86 lbs. absolute = 316·9° Temperature.
Dryness = ·978.

First set off on the entropy diagram (59), at the 400° temperature level, ·978 of the length between the water line and saturated line, and drop a vertical for adiabatic expansion. Assume that the turbine expansion line, due to the reheating effects of blade friction losses, occupies a position, say, ·4 of the distance from the adiabatic line to the saturation line at the 316·9 temperature level. Mark this point and draw in the turbine expansion line as shown in the sketch. Next measure the entropy at the two temperature levels, which give ·952 at 400° temperature and 1·087 at 316·9° temperature.



No. 59.—Entropy Diagram.

Then, B.T.U. drop = Mean Entropy × Difference of Temperature.

Therefore, $\phi = \left(\frac{\cdot 952 + 1\cdot 087}{2}\right) \times (400 - 316\cdot 9) = 84\cdot 67$ **B.T.U.**

Assume that nozzle velocity friction loss = 10 per cent. giving nozzle efficiency as 90 per cent. or ·90.

Assume that blade velocity friction loss = 12 per cent., giving blade efficiency as 88 per cent. or ·88.

Then,

Nozzle calculated exit velocity $= \sqrt{84{\cdot}67 \times 64{\cdot}4 \times 778} = 2059$ **ft. per sec.**

Actual velocity $= 2059 \times {\cdot}9 = 1853$ **ft. per sec.**

Blade speed $= \dfrac{6 \times 3{\cdot}1416 \times 593}{60} = 187$ **ft. (nearly) per sec.**

Note.—Mean diameter of blades = 6 ft. Revolutions, 593 per minute.

The velocity diagram is then set off exactly as described on p. 384, but in the present case the blade friction-velocity loss is taken as 12 per cent., equal to a blade efficiency of 88 per cent.

The successive relative and absolute velocities are, therefore, each equal to ·88 of the one before.

Take 1st moving blade relative exit velocity :—

Then, $1673 \times {\cdot}88 = 1472$ **ft.**

Take 1st fixed blade absolute exit velocity :—

Then, $1300 \times {\cdot}88 = 1144$ **ft.**

Each velocity is developed in the same manner.

The actual blade angles are shown as compared with the calculated blade angles, and it will be noticed that in every row the actual admission angle is in excess of the calculated angle.

Work Done in Stage.

Work done in B.T.U. $= \dfrac{\text{Total Sum of difference of squares of velocities}}{64{\cdot}4 \times 778}.$

Then,

$$1853^2 - 1673^2 = 635000 \text{ ft.}$$
$$1472^2 - 1300^2 = 477000 \text{ ,,}$$
$$1144^2 - 973^2 = 361400 \text{ ,,}$$
$$856^2 - 687^2 = 260700 \text{ ,,}$$
$$604^2 - 443^2 = 168500 \text{ ,,}$$
$$389^2 - 227^2 = 99780 \text{ ,,}$$
$$199^2 - 80^2 = 33210 \text{ ,,}$$
$$70^2 - 145^2 = -16130 \text{ ,,}$$

Note.—The last velocity extraction gives a *minus* result which is deducted from the total as shown.

$$\text{B.T.U.} = \frac{635000+474000+361400+260700+168500+99780+33210-16130}{64{\cdot}4 \times 778} = 40{\cdot}24.$$

Stage Efficiency $= 40{\cdot}24 \div 84{\cdot}67 = {\cdot}475.$

or, ${\cdot}475 \times 100 = 47{\cdot}5$ **per cent.**

Note.—$84{\cdot}67 =$ **B.T.U. drop from entropy diagram calculations.**

— 95 „ „ = 324

Note.—84·67 = B.T.U. drop from entropy — [illegible] relations.

Indicated Work.

By this is meant the actual heat units or foot-lbs. of energy converted into useful work, and the amount can be found by any of the following methods, the blade friction losses, etc., being known or assumed :—

Method 1.

Indicated Work in B.T.U. per lb. = Available Heat — Heat expended in blade friction, leakage, and carry-over.

Method 2.

Indicated Work in B.T.U. per lb.

$$= \frac{\text{Distance moved in feet} \times \text{Force exerted by steam in lbs}}{778}.$$

Method 3.

Indicated Work in B.T.U. per lb.

$$= \frac{\text{Sum of differences of square of steam velocities}}{64\cdot4 \times 778}.$$

And, Efficiency = Work done ÷ Heat (or velocity) supplied.

The following working shows an example of each method applied to the same case.

Example.—The following worked out examples illustrate how the work done by the steam can be estimated from the velocity triangles, and from this the turbine efficiency determined.

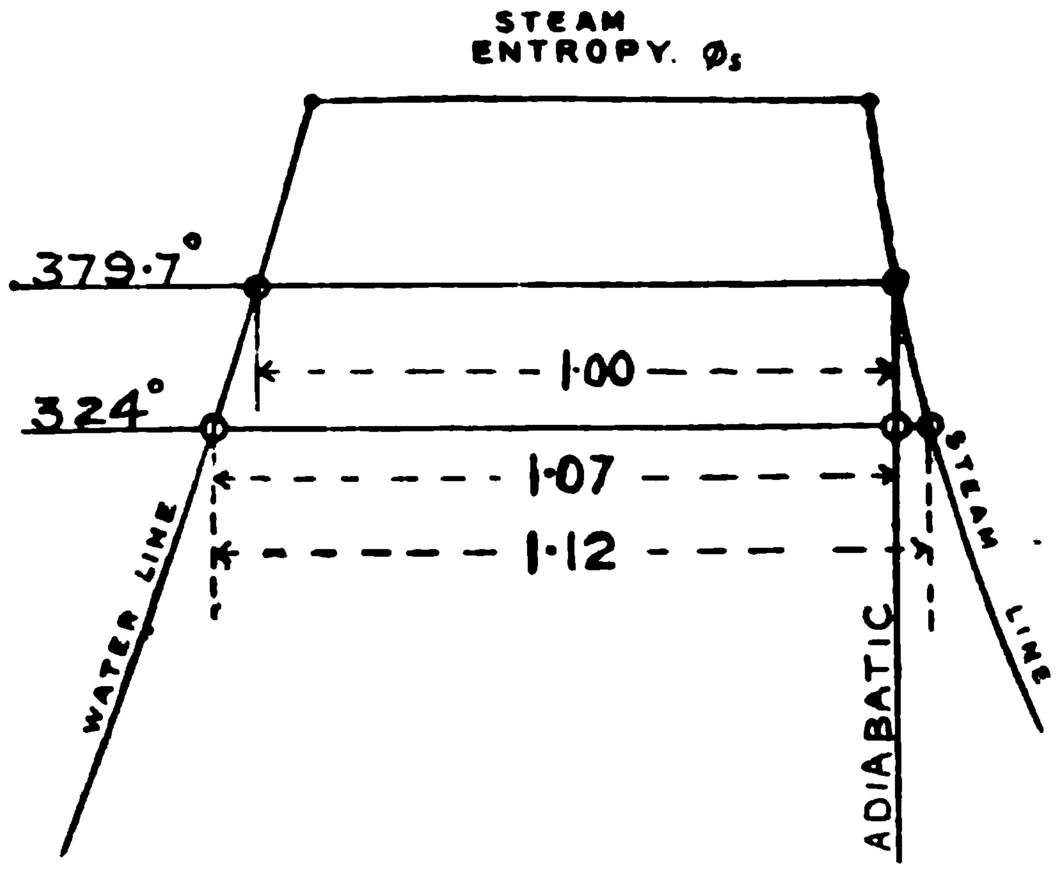
Data.—

Initial nozzle steam 180 lbs. gauge.
Exit nozzle steam 80 „ „
Dryness of Initial Steam = 1.
Adiabatic expansion assumed.
Nozzle angle = 20°.
Nozzle Efficiency = ·81.
Blade Velocity Efficiency = ·88.
Blade Speed = 150 ft. per sec.
First stage consists of three rows of moving blades and two rows of fixed blades.
Horse Power, 6000.
Revolutions, 450 per min.
Estimated consumption = 14 lbs. steam per horse power per hour.

Note.—The reheating effect of blade friction loss is neglected in the following calculations :—

Available Heat.

180 + 15 = 195 lbs. absolute = 379·7°
80 + 15 = 95 „ „ = 324°



No. 61.—Entropy Diagram.

Draw horizontal lines across the entropy diagram (61) at the two temperature levels of 379·7° and 324° dropping a vertical line for adiabatic expansion, then measure the entropy, as shown, which will be found to be 1·005 and 1·070 respectively.

Then, Mean Entropy $= \dfrac{1{\cdot}00 + 1{\cdot}070}{2} = 1{\cdot}035.$

Temperature Difference $= 379{\cdot}7^\circ - 324^\circ = 55{\cdot}7.$

So that, B.T.U. set free $= 1{\cdot}035 \times 55{\cdot}7 = 57{\cdot}6.$

Nozzle Steam Velocity $= \sqrt{57{\cdot}6 \times 778 \times 64{\cdot}4 \times {\cdot}81} = 1526$ ft. per sec.

Dryness of steam $= 1{\cdot}07 \div 1{\cdot}12 = {\cdot}955.$

Specific Volume of Steam at 80 lbs. absolute (From Table) $= 5{\cdot}348$ cub. ft.

Actual Volume at mouth $= 5{\cdot}348 \times {\cdot}955 = 5{\cdot}10$ cub. ft.

Velocity Triangles (62).

In allowing for blade friction notice that for each moving blade row the relative exit velocity (V_3) is equal to ·88 of the previous relative admission velocity.

Example 1.—At 1st moving blade row relative admission velocity = 1387, then, 1387 × ·88 = 1220 for relative exit velocity.

Also note that the absolute exit velocity from each fixed blade row is equal to ·88 of the previous admission velocity.

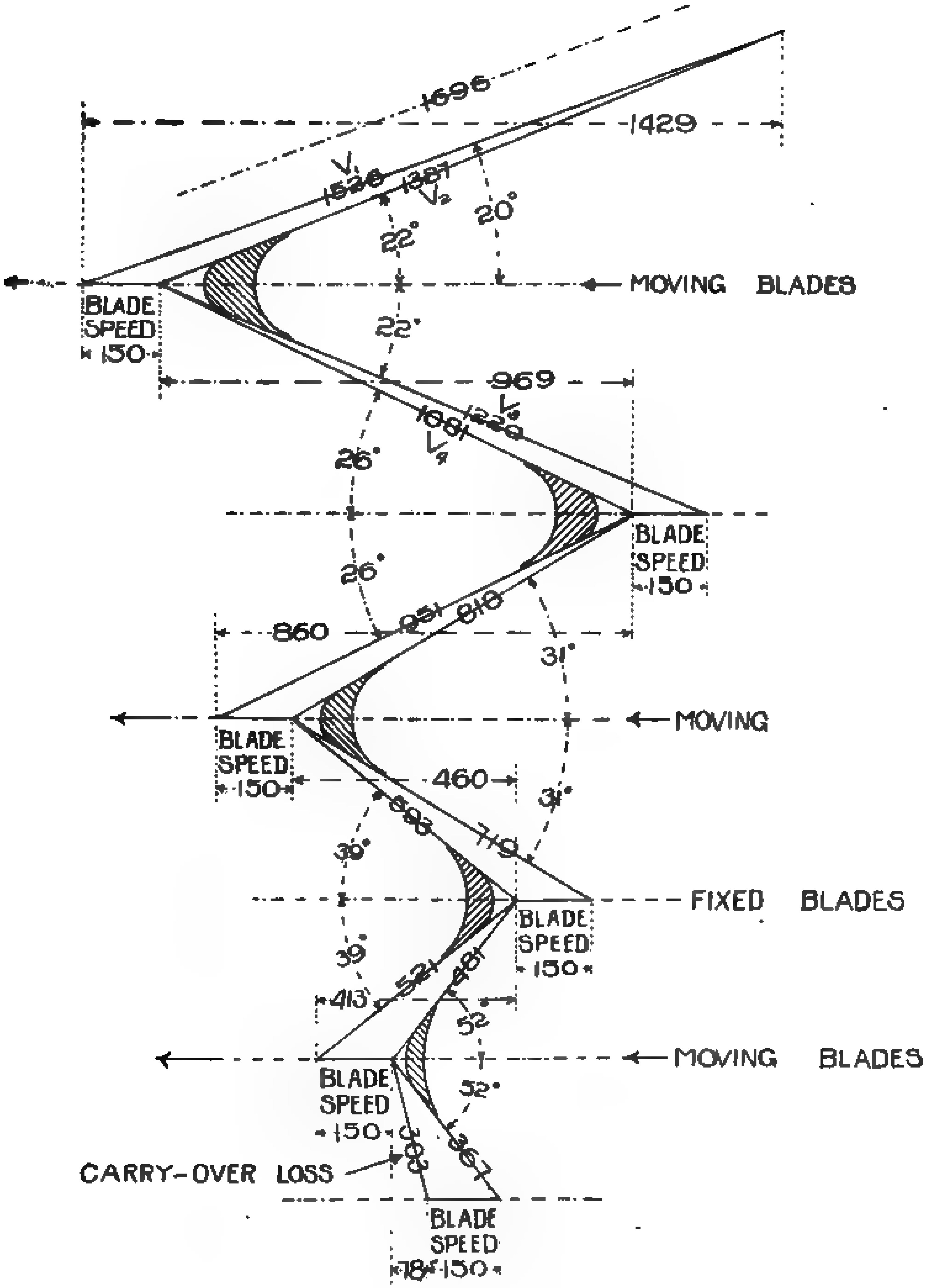
Example 2.—At 1st fixed blade row (62) the admission velocity= 1081, then, 1081 × ·88 = 951 for absolute exit velocity which is at the same time the absolute admission velocity for the next moving blade row.

Also,

$818 \times {\cdot}88 = 719,$

$593 \times {\cdot}88 = 521,$

$418 \times {\cdot}88 = 367.$



No. 62.—Diagram of Velocity Triangles.

For Impulse Turbine.

[*To face page* 416.

As the steam velocity steadily falls throughout the stage, it will be evident that the blade angles automatically increase, as shown in the diagram, the final blade angle being 52° in the present example in place of 20° at the initial end of the stage.

Indicated Work = Available Heat in B.T.U. — Heat expended in blade friction, carry-over, &c.

Method 1.—

Available Heat = 57·6 B.T.U.

$$\text{Heat expended in blade friction and carry-over}\left\{\begin{array}{lll} \text{1st Row} & 1387^2 - 1220^2 = & 436000 \\ \text{2nd} \;\; ,, & 1081^2 - \;\; 951^2 = & 264600 \\ \text{3rd} \;\; ,, & 818^2 - \;\; 719^2 = & 152300 \\ \text{4th} \;\; ,, & 593^2 - \;\; 521^2 = & 80400 \\ \text{5th} \;\; ,, & 418^2 - \;\; 367^2 = & 40100 \\ \text{Carry-over loss} = & 303^2 = & 91790 \end{array}\right.$$

Then,

$$\text{Total loss} = 436000 + 264600 + 152300 + 80400 + 40100 + 91790 = 1065190.$$

$$\text{B.T.U.} = \frac{1065190}{64{\cdot}4 \times 778} = 21{\cdot}26.$$

$$\text{Indicated Work} = (57{\cdot}6 \times {\cdot}81 - 21{\cdot}26) = 25{\cdot}39 \text{ B.T.U.}$$

$$\text{Efficiency} = 25{\cdot}29 \div 57{\cdot}6 = {\cdot}44, \text{ or } 44 \text{ per cent.}$$

Note.—64·4 = 32·2 × 2 gravity acceleration constant.
778 ft.-lbs. = 1 B.T.U.

Method 2.—

$$\text{Indicated Work in B.T.U.} = \frac{\text{Distance moved in feet} \times \text{Force in lbs.}}{778}$$ (see p. 388).

Referring to the velocity diagram (62), it will be seen that the change of velocity at the first moving blades = 1429 + 969 = 2398 ft., at the second moving blades = 860 + 460 = 1320, and at the third = 413 + 78 = 491. The distance moved in feet in each case = 150 ft.

$$\text{Then,} \quad \frac{(2398 \times 150) + (1320 \times 150) + (491 \times 150)}{32{\cdot}2 \times 778} = 25{\cdot}2 \text{ B.T.U.}$$

$$\text{Efficiency} = 25{\cdot}2 \div 57{\cdot}6 = {\cdot}44 \text{ (nearly).}$$

Note.—The slight difference in the result is due to the approximate method of measuring the velocities by scale.

Method 3.—

$$\text{Indicated Work} = \frac{\text{Sum of difference of square of steam velocities from diagram.}}{64{\cdot}4 \times 778}$$

Notice that the *steam* velocity, absolute and relative, of each triangle is first squared, then subtracted, and the total sum taken.

$$\text{Indicated Work}\left\{\begin{array}{l} 1526^2 - 1387^2 = 403000. \\ 1220^2 - 1081^2 = 320000. \\ \;\; 951^2 - \;\; 818^2 = 235200. \\ \;\; 719^2 - \;\; 593^2 = 165100. \\ \;\; 521^2 - \;\; 418^2 = 96600. \\ \;\; 367^2 - \;\; 303^2 = 42910. \end{array}\right.$$

$$\text{Then,} \quad \frac{403000 + 320000 + 235200 + 165100 + 96600 + 42910}{64{\cdot}4 \times 778} = 25{\cdot}2 \text{ B.T.U.}$$

$$\text{Efficiency} = 25{\cdot}2 \div 57{\cdot}6 = {\cdot}44 \text{ (nearly).}$$

Nozzle Openings.

Throat Pressure $= 195 \times \cdot 58 = 113$ lbs. absolute.
113 lbs. absolute $= 336 \cdot 7$ temperature and $3 \cdot 86$ specific volume (from Table).

Referring to the entropy diagram, draw a line across at the level of 336·7 temperature, and measure from the water line to the vertical representing adiabatic expansion, also measure across to the saturation line. These measurements, representing entropy, will be found equal to 1·06 and 1·1.

Then, Dryness $= 1 \cdot 06 \div 1 \cdot 1 = \cdot 963.$

Actual Steam Volume $= \cdot 963 \times 3 \cdot 86 = 3 \cdot 717$ cub. ft.

$$\text{Mean Entropy} = \frac{1 \cdot 005 + 1 \cdot 06}{2} = 1 \cdot 032.$$

Temperature Difference $= 379 \cdot 7 - 336 \cdot 7 = 43.$

Drop in B.T.U. $= 43 \times 1 \cdot 032 = 44 \cdot 37.$

Throat Velocity $= \sqrt{44 \cdot 37 \times 64 \cdot 4 \times 778 \times \cdot 81} = 1342$ ft. per sec.

$$\text{Steam Flow per sec.} = \frac{\overset{\text{H.P.}}{6000} \times 14}{60 \times 60} = 23 \cdot 33 \text{ lbs.}$$

Volume per sec. $= 23 \cdot 33 \times 3 \cdot 717 = 86 \cdot 71$ cub. ft.

$$\text{Throat Area} = \frac{86 \cdot 71 \times 144}{1342} = 9 \cdot 3 \text{ sq. in.}$$

Assume 10 nozzles.

$$\text{Then, Area per nozzle} = \frac{9 \cdot 3}{10} = \cdot 93 \text{ sq. in.}$$

Assuming square section at throat $\sqrt{\cdot 93} = \cdot 96$ in. sides.

$$\text{Mouth area} = \frac{23 \cdot 33 \times 5 \cdot 1 \times 144}{1526} = 11 \cdot 22 \text{ sq. in.}$$

$$\text{Area of each nozzle (10 fitted)} = \frac{11 \cdot 22}{10} = 1 \cdot 122 \text{ in.}$$

Depth of nozzles = say, $1\frac{1}{8}$ in.

Then, Width of nozzles $= 1 \cdot 122 \div 1 \cdot 125 = \cdot 99$, say, 1 in.

Width of Mouth $= 1$ in. $\div \cdot 342 = 2.9$ in.

Note.—1 in. $\div$ sine of angles $20^\circ = 1 \div \cdot 342 = 2 \cdot 9$ in.

Blade Heights (see page 407).

Allow ·9 increase of area through blades to allow for thickness of blade metal, which latter reduces the effective opening area for steam flow.

$$\text{Height of 1st Moving Blades} = \frac{23 \cdot 33 \times 5 \cdot 1 \times 144}{10 \times 2 \cdot 9 \times 1081 \times \cdot 374} = 1 \cdot 46 \text{ in.}$$

Note.—Sine of angle 22' = ·374.

Actual Height $= 1{\cdot}46 \div {\cdot}9 = 1{\cdot}62$ **in.**

Height of 1st Fixed Blade $= \dfrac{23{\cdot}33 \times 5{\cdot}1 \times 144}{10 \times 2{\cdot}9 \times 818 \times {\cdot}438} = 1{\cdot}65$ **in.**

Or, $\dfrac{1081 \times 1{\cdot}46 \times {\cdot}374}{818 \times {\cdot}438} = 1{\cdot}65$ **in.**

Note.—The sine of angle 26° = ·438 (from Table, p. 442).

Actual Height $= 1{\cdot}65 \div {\cdot}9 = 1{\cdot}83$ **in.**

Height of 2nd Moving Blade $= \dfrac{23{\cdot}33 \times 5{\cdot}1 \times 144}{10 \times 2{\cdot}9 \times 593 \times {\cdot}515} = 1{\cdot}93$ **in.**

Actual Height $= 1{\cdot}93 \div {\cdot}9 = 2{\cdot}14$ **in.**

Note.—Sine of angle 31° = ·515.

Height of 2nd Fixed Blade $= \dfrac{23{\cdot}33 \times 5{\cdot}1 \times 144}{10 \times 2{\cdot}9 \times 418 \times {\cdot}629} = 2{\cdot}25$ **in.**

Actual Height $= 2{\cdot}25 \div {\cdot}9 = 2{\cdot}5$ **in.**

Note.—The sine of angle 39° = ·629.

Height of 3rd Moving Blade $= \dfrac{23{\cdot}33 \times 5{\cdot}1 \times 144}{10 \times 2{\cdot}9 \times 303 \times {\cdot}788} = 2{\cdot}48$ **in.**

Actual Height $= 2{\cdot}48 \div {\cdot}9 = 2{\cdot}75$ **in.**

Note.—Sine of angle 52° = ·788.

The 1st stage blade heights therefore work out as follows:—

		Blade Height Increase.
1st Row Moving - - - - -	1·62 in.	0 in.
2nd „ Fixed - - - - -	1·83 in.	·21 in.
3rd „ Moving - - - - -	2·14 in.	·31 in.
4th „ Fixed - - - - -	2·5 in.	·36 in.
5th „ Moving - - - - -	2·75 in.	·25 in.

In practice the blade angles at exit edges are *less* than at admission; this is designed to allow for the reduced thickness of steam flow which takes place through the blade openings owing to a "spreading" action of the jet. The sketches (page 432-433) shows the actual angles as adopted in a modern type Curtis turbine.

SECTION VIII.

DESCRIPTION OF THE CURTIS TURBINE, ETC.

THE Curtis turbine is double compounded, that is, compounded for pressure and also for velocity. The pressure compounding is arranged for by the successive stage nozzles, and the velocity compounding by the number of rows of fixed and moving blades (often

No. 63.—Curtis Type Impulse Turbine Nozzle Blades and Control Valve.*

In this case the nozzle control valves are of flat face construction, and slide up or down by means of a screw on the spindle operated by hand.

termed "buckets") on each stage wheel. The steam, in expanding through the nozzles, changes the internal energy contained into velocity, and in passing through the moving blades or buckets this velocity is utilised in the form of mechanical work exerted on the rotor.

The De-Laval type (see p. 13) is an example of a single wheel impulse turbine, consisting of a set of nozzles placed round the

* Reprinted from *Engineering*.

circumference, and a single ring of buckets on the wheel. As mentioned elsewhere, for maximum efficiency the blade speed should be equal to half that of the steam, and as the steam speed is seldom less than 3000 ft. per sec., the blade speed would require to be about 1500 ft per sec., which is, needless to say, quite out of

No. 64.—Intermediate Stage Impulse Wheel, showing Method of Connection to Arms and to Shaft.*

(U.S. Destroyers, "Perkins" and "Sterrett."

The hub is secured to the shaft by keys, and is turned $\frac{1}{1000}$ in. less in diameter than the shaft before being forced on by hydraulic pressure.

the question for marine practice.

In the Curtis turbine this difficulty is overcome by arranging for more than one set of nozzles, and allowing only a limited pressure drop in each, which results in keeping down the exit velocity of the steam from each successive set of nozzles, and of course the blade speed is reduced in the same ratio. To further reduce the steam velocity, from three to four rows of moving blades are fitted on each

wheel, and two or three rows of fixed blades to the cylinder, there being usually one row less of fixed blades than of moving blades.

The steam flow passages or openings between the blades are generally parallel (see sketch 47), and the pressure remains nearly constant throughout a stage.

In the moving rows of blades velocity energy is extracted, so that the steam speed falls from row to row.

In the fixed blade rows the steam flow is merely redirected and

SECTION ON LINE A.A.

No. 65.—Method of Holding in Place the Fixed Blades of the Intermediate Wheel Stages to the Casing.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

The above maintains from the 2nd to the 6th stage, as each of these stages consist of 3 moving blade rows and 2 fixed blade rows.

no work extracted, so that the heat conditions remain the same, less that used up in overcoming blade friction.

Each row of moving blades takes out of the steam a velocity equal to about twice that of its own velocity, and in this way the speed of the steam is continuously reduced from the initial end to the exhaust end of the turbine.

Curtis turbines are different from the Parson's type by being "single cylinder," that is, the complete range of expansion from boiler steam to condenser pressure is generally carried out in the one turbine, instead of being arranged on the H.P. and two L.P. turbine

* Reprinted from *Engineering*.

system of Parson's, although in the case of large powers more than one turbine requires to be fitted.

Construction.

The turbine consists of two principal parts, the rotor or moving

No. 66.—Section of Drum Stage, showing Rim and Moving Blade Rows.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

No. 67.—Details of Intermediate Impulse Stage.*

(2nd to 6th Stages.)

(U.S. Destroyers, "Perkins" and "Sterrett.")

Observe that the wheel is rivetted to the steel plate discs on either side.

part, and the cylinder (sometimes called the "stator") or stationary part.

The cast-steel *cylinder* includes the steam chests or nozzle boxes, the diaphragms or divisions between the different stages, and the fixed blade rows.

No. 68.—Fixed Blade Holder and Nozzle of 1st Drum Stage.*
(U.S. Destroyers, "Perkins" and "Sterrett.")

The blade on right is the last fixed one of stage 7, and the blade on left the first fixed one of stage 8; the slot at *g* allows of flexibility to the steam leakage baffle serrations in case of actual metallic contact with the revolving wheel rim.

No. 69.—1st Stage Moving Blade Rows.*
(U.S. Destroyers, "Perkins" and "Sterrett.")

Observe the stepping of the wheel rim and of the blades. Notice that the blade lengths require to be increased at each row, not for increase of steam volume, but for decrease of steam velocity

* Reprinted from *Engineering*.

The *rotor* is made up as follows:—A hollow steel shaft on which are forced by hydraulic pressure the cast-steel wheels, which are afterwards secured by keys, the shaft being stepped at each wheel.

No. 70.—Diaphragm Division Plates between Impulse Stages.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

Discs of rolled-steel plate are bolted or riveted to the wheels, and the rims on which the moving blades are fixed are also bolted or riveted to the discs.

The diaphragms or divisions (70) split up the turbine longitudinally into stages, each stage consisting of a set of nozzles and a companion

set of moving and fixed blades, the moving blades being secured to the wheel of that stage.

The diaphragms are made of steel plate or of cast steel, and are fitted with a grooved bush (71) round the shaft opening to reduce steam leakage from one stage to another.

The bush referred to is of bronze, and the grooves are of V section to reduce danger of accident by contact. A certain amount of leakage is of course unavoidable, as a working clearance must be allowed for.

No. 71.—View of Diaphragm Plate with Steam Baffle Serrations at Hub.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

Steam leakage from stage to stage is reduced by means of the serrations formed on the diaphragm hub bush, which is also to a slight extent flexible in action by being arranged in an overhung position.

Nozzles.

The nozzles are rectangular in section, and all except the first stage are usually of the non-expanding type, that is, the area at the mouth is equal to that at the throat: at the first stage the nozzle is of the expanding type. The openings are rounded off by easy fillets, and the depth of nozzle at the mouth is regulated by height of the first moving blade rows. As stated before, in the nozzles the steam falls in pressure but increases in velocity; pressure energy being converted into velocity, and the number of heat units converted in this way is called the "heat drop" in the nozzle. Some of the heat is used up in overcoming friction, otherwise the expansion is "adiabatic," that is, the work done on the steam to increase its speed is balanced by a corresponding loss of heat.

The nozzle angle is 20° at the first stages and 25° at the later stages: the nozzles occupy the arc of a circle, and deliver the steam

* Reprinted from *Engineering.*

directly on to the first row of moving blades, which are in most cases about 1¼ in. in height at the first stage.

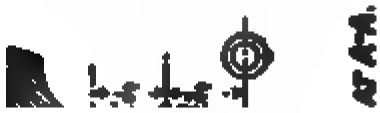
The nozzle discharge openings form a continuous area, less that

No. 72.—View of Nozzle Openings to which the Nozzle Plates are Screwed.*
(U.S. Destroyers, "Perkins" and "Sterrett.")

due to the required thickness of metal separating each nozzle from its neighbour; it will therefore be obvious that the head of steam flows in the form of a solid stream, the depth of which is equal to that of the blade rows, and the width of which is equal to the circumferential length of the combined nozzle discharge openings.

Adjacent nozzles should always be opened for any given power, otherwise losses by side eddy currents would result.

The circumferential arc occupied by the first stage nozzles is



LOOKING AFT.

No. 73.—Nozzle Plate Seatings on Casings.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

The nozzle plates are screwed on to the casings at the position shown in the illustration. On the right a sectional view through the nozzles is also shown, with dimensions.

* Reprinted from *Engineering*.

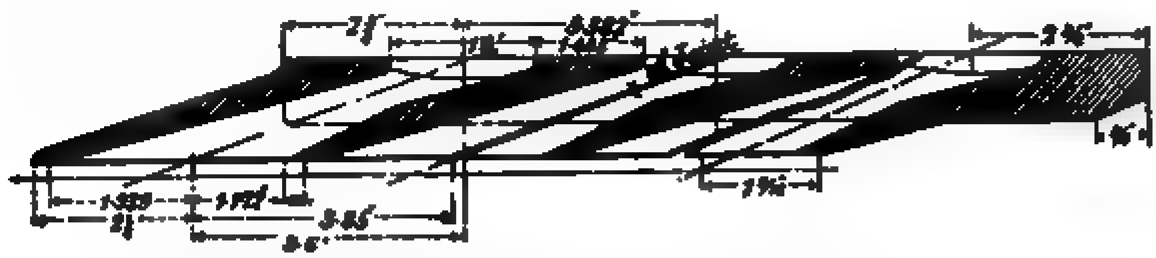
about equal to 45°, but as the stages progress the length of arc increases, and from the fifth stage onwards the nozzle openings extend right round the circumference, and in addition the nozzle opening areas require to be increased and the blade heights

Astern

No. 74.—Astern Turbine Nozzle Inlet Openings.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

At the initial stages the fixed blades only extend slightly beyond the nozzle extremities on either side.



No. 75.—Astern Turbine Nozzle Openings.*

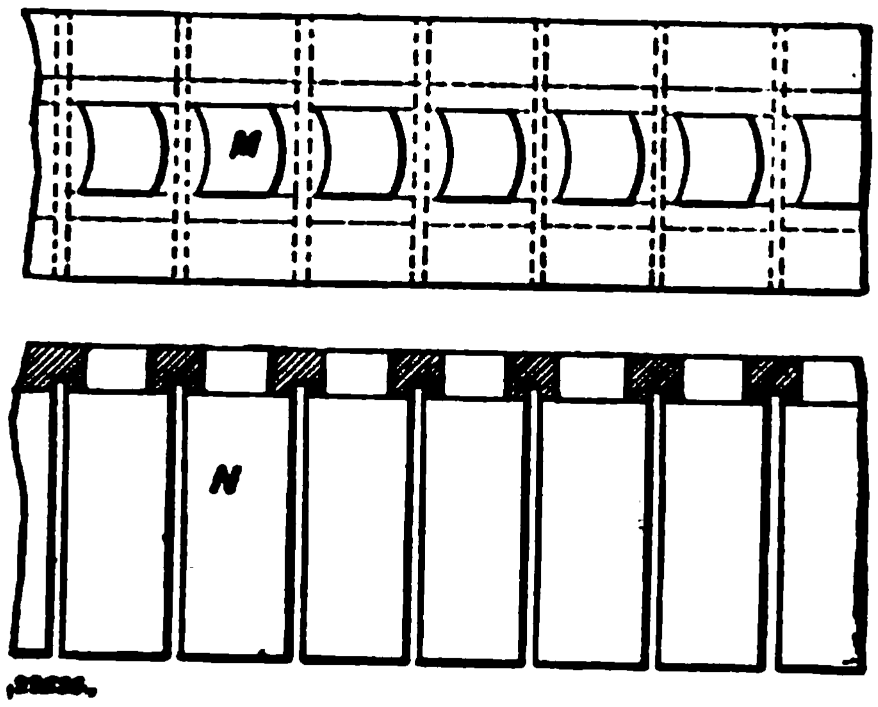
(U.S. Destroyers, "Perkins" and "Sterrett.")

Notice the large amount of taper required on the nozzle passages to allow for a considerable range in expansion.

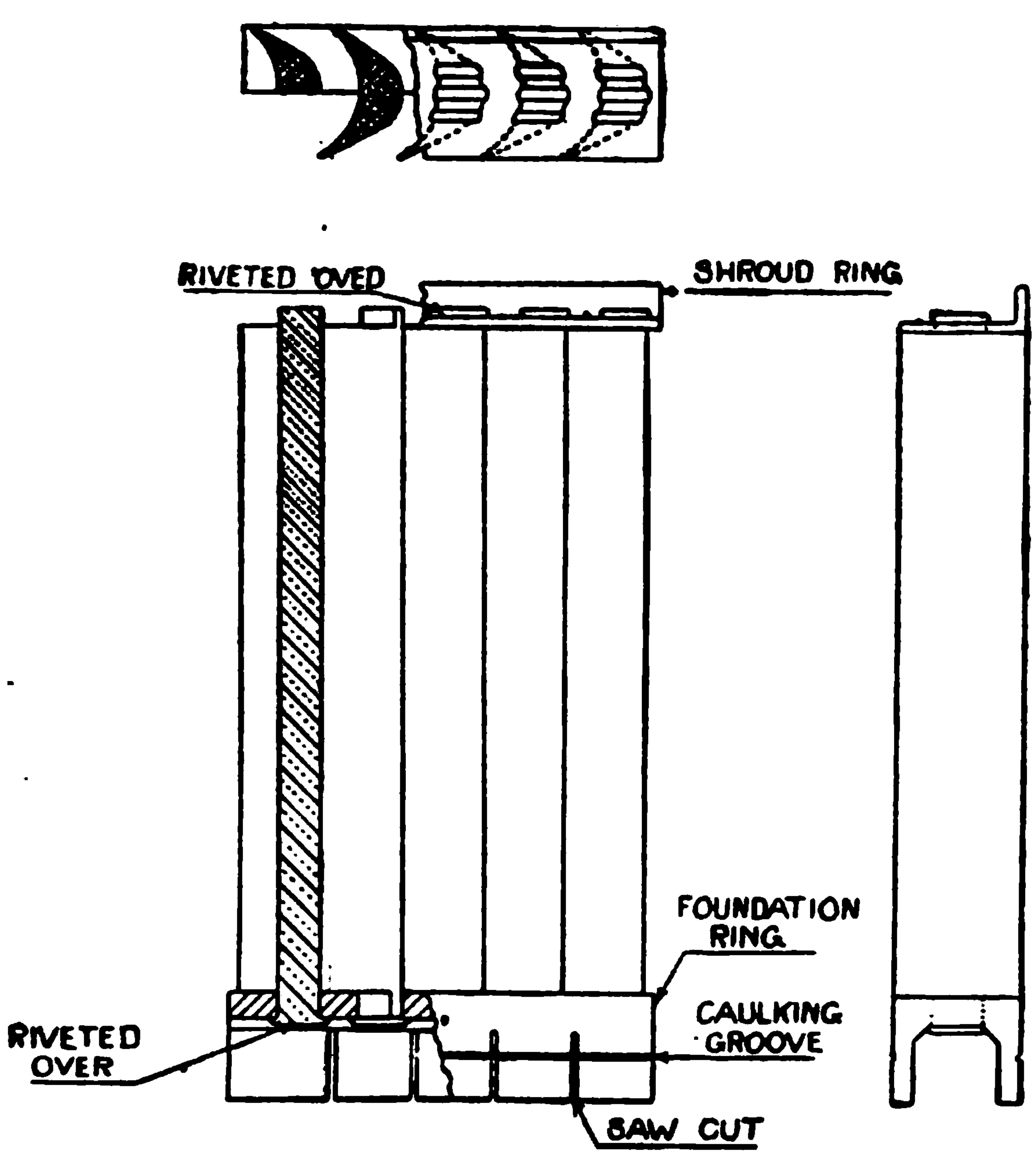
No. 76.—View of Drum Stage Nozzle Openings.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

In the drum portion of the turbine the steam admission takes place right round the circle, and this is arranged for by fitting 8 separate nozzle segments, one of which is shown in the lower view.



No. 77.—Plan of Blades (M), also Elevation of Foundation Ring (N) showing Saw-cuts to allow for Bending to Shape.*

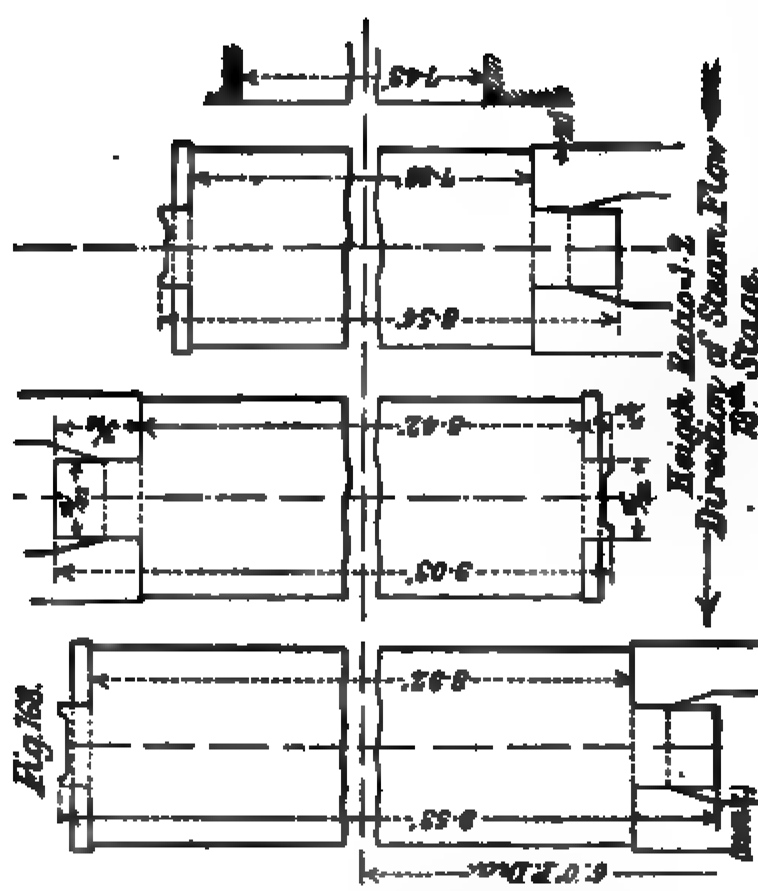


No. 78.—Curtis Type Blading with Foundation Ring.

* Reprinted from *Engineering*.

lengthened to allow for the rapid increase of steam volumes at the lower pressures.

In any stage the pressure is practically constant from end to



No. 79.—13th and 14th Stage Impulse Blading.*

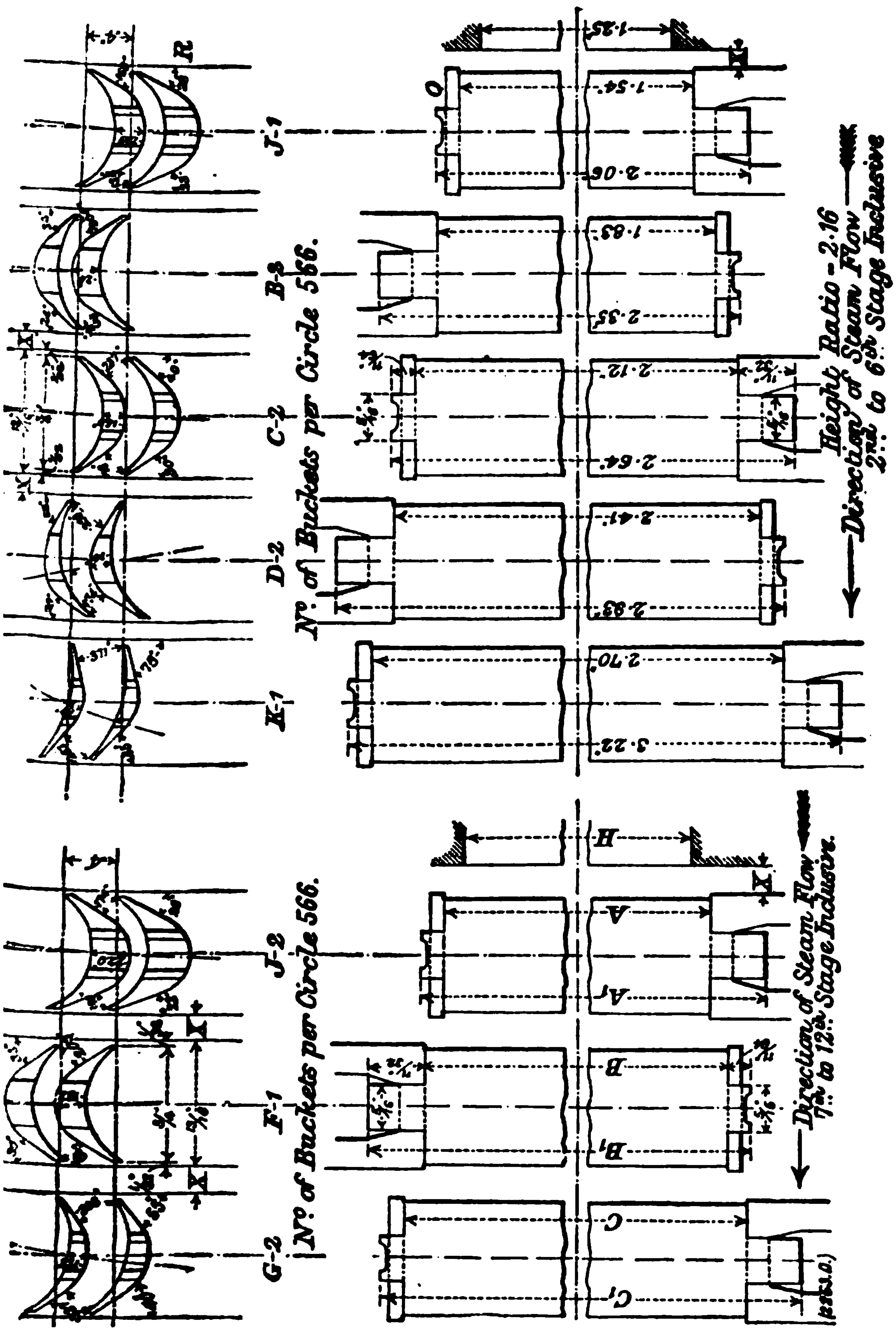
(U.S. Destroyers, "Perkins" and "Sterrett.")

Both of these stages consist of 2 moving rows and 1 fixed row, the blades of which are all wider than those of the previous stages. Notice that the blades are canted, thus giving an increase to the admission angle and a decrease to the exit angle.

end, but it should be noted that the working steam travels directly through the various blades in a spiral path. Holes are cut in the plates of the disc wheels to allow of equal pressure on either side.

Blades.

The blading material employed is bronze, first produced in long



No. 80.—2nd to 12th Stage Impulse Blading.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

It should be noted that the blades are of less thickness and width than those in the previous stages, but as before the blades are angled axially. Up to the 6th stage, 3 moving rows and 2 fixed rows are fitted; beyond this, each stage consists of 2 moving rows and 1 fixed row.

* Reprinted from *Engineering*.

bars of the correct section, which are then cut up into the required blade lengths, the ends being milled to shape.

The blades are next put together in section lots on to a steel channel base ring, and the ends of the blades riveted over the ring. The channel or foundation ring is sawed out at regular intervals to allow of flexibility when being bent over to fit on the rim of the wheel.

When the blades are fitted in position on the rim the channel base is caulked in place, and a shroud ring put over the blade ends, which are then riveted over.

The stationary blades are fixed in a similar manner to that of the moving blades.

The blades differ in section throughout a stage, becoming flatter in shape and wider in circumferential pitch as the stage progresses to allow for the increase in steam volume and the reduced steam speed. The blades are also made longer and much thinner in section at the low pressure stages for the same reasons, and the angles increased.

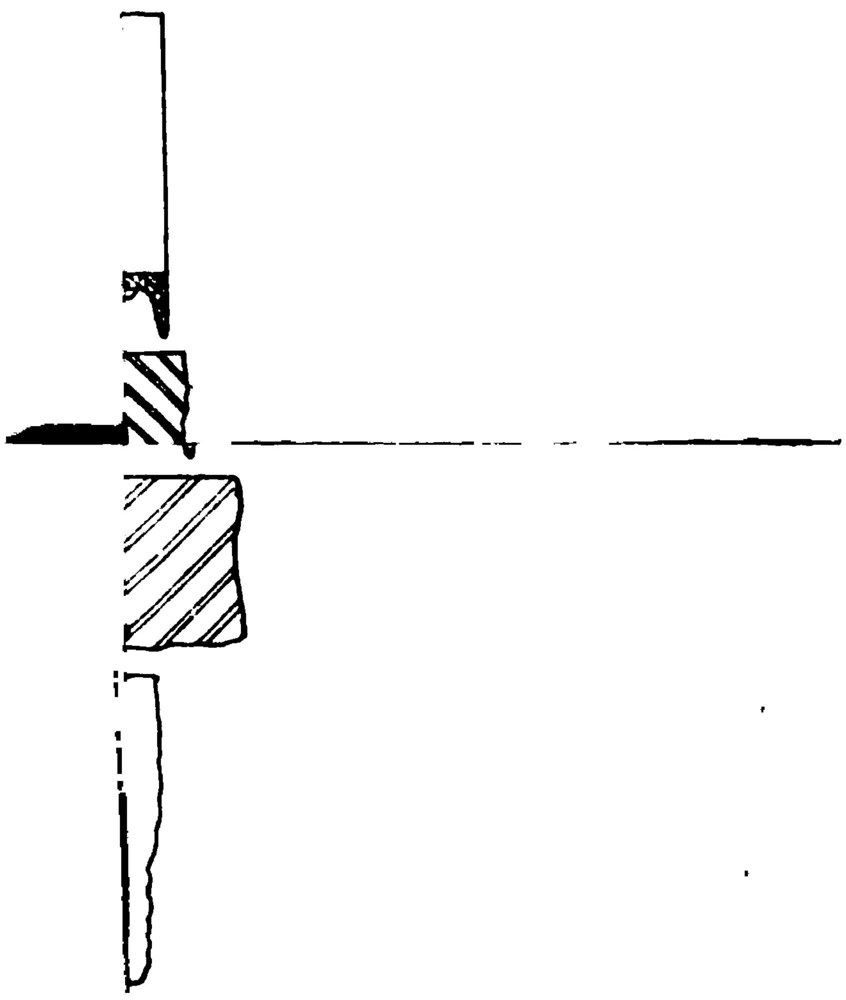
Effective Blade Height.

The clear distance between the shroud ring and the top edge of the foundation ring of a Curtis type blade is known as the "effective height," this being the clear area open for steam flow.

No. 81.
C, Tip Clearance.
E, Effective Height.

Action of the Steam.

Steam enters the nozzle box at a few pounds less than boiler pressure, and passing through the nozzles falls in pressure but acquires velocity, the increase in velocity being dependent on the "heat drop" occurring in the nozzle.



ion Blading.

›f drum.

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steam flow away
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an actual "knife
lades come into
; over or grinding
of damage done
otor and casing
clearance, is a

Leaving the nozzle openings the steam strikes the first row of moving blades or buckets, giving up a part of its velocity, and issues in a solid jet against the first row of fixed or guide blades, which deflect the flow into the correct striking angle for the second row of moving blades; again the steam parts with a portion of its velocity, and, as before, is redirected by means of a second row of guide blades to the third row of moving blades, and so on throughout that particular stage.

At the second stage the same process is repeated with the steam at reduced pressure, and the same for all following stages, the steam pressure falling from one stage to the next, but remaining practically constant throughout any one stage.

Blade friction reduces the steam velocity at each row of blades, but the heat temporarily lost in this way reappears later on as, what is termed, "regenerated heat."

If the relative steam velocity at entrance is, say, 1000 ft. per sec., and the blade friction loss 12 per cent., then the actual relative steam velocity at exit $= 1000 \times \frac{88}{100} = 880$ ft. as the efficiency $= 100 - 12 = 88$ per cent. This obtains at all the blade rows, and reduces the steam velocities all the way. If no friction loss took place the relative steam speed at exit would equal that at admission.

Pressures.

In the Curtis turbine the initial pressure is usually much above that employed in the Parson's type, the steam usually being at a pressure of about 250 lbs. or more, but the pressures on the blading is less than that of the Parson's owing to the pressure drop in the nozzles.

Pressure Drops per Stage.

Taking a turbine with, say, seven stage expansions, the pressures at each stage would be somewhat as follows.

	Nozzle Pressure at Admission (absolute)	Throat Pressure (absolute)	Pressure at Stage (absolute)
Stage 1 - -	265	153	79
„ 2 - -	79	45·6	41·7
„ 3 - -	41·7	24	21·2
„ 4 - -	21·2	12·2	10·4
„ 5 - -	10·4	6	4·9
„ 6 - -	4·9	2·83	2·2
„ 7 - -	2·2	1·27	1

Note.—Each throat pressure is equal to the nozzle initial pressure +·577 (see p. 373).

It will be noticed that the pressure in any stage is also the nozzle initial pressure for the next stage.

Example.—

Stage 1. Then, 265 × ·577 = 152·9, say, 153 lbs.
" 2. " 79 × ·577 = 45·58, say, 45·6 lbs.

Drum Stages.

The Brown-Curtis turbine (82) differs from the ordinary Curtis turbine by the addition of a tapered drum, which is placed on the lower pressure end of the usual wheel stages, and has a series of impulse stages (usually seven or eight), or (in recent designs) reaction stages, with two or three moving blade rows and one or two fixed blade rows to each stage. The drum is stepped down and the casing stepped up to allow for the required increase of blade heights at this position of the turbine, and it will be noted, the mean diameter across blades is thus kept constant.

With impulse blading at the drum stages the nozzle openings extend round the full circumference of the casing. The diaphragms between each stage form rings, and these diaphragms are fitted with a minimum working clearance round the shaft to reduce steam leakage between the stages.

With reaction blading, each stage or "expansion" consists of 1 blade row, followed by one 3 or 4 blade rows, turn about throughout the stage.

The drum consists of a cast steel hub with arms, the drum being secured to these by means of steel plates; the forward plate serves to shut off the steam pressure from the preceding stage, and also acts as a dummy piston by means of the pressure load exerted on the plate area, which is designed to balance the propeller thrust; the after plate is also supplied with a series of holes which puts the inside of the drum in communication with the terminal exhaust pressure.

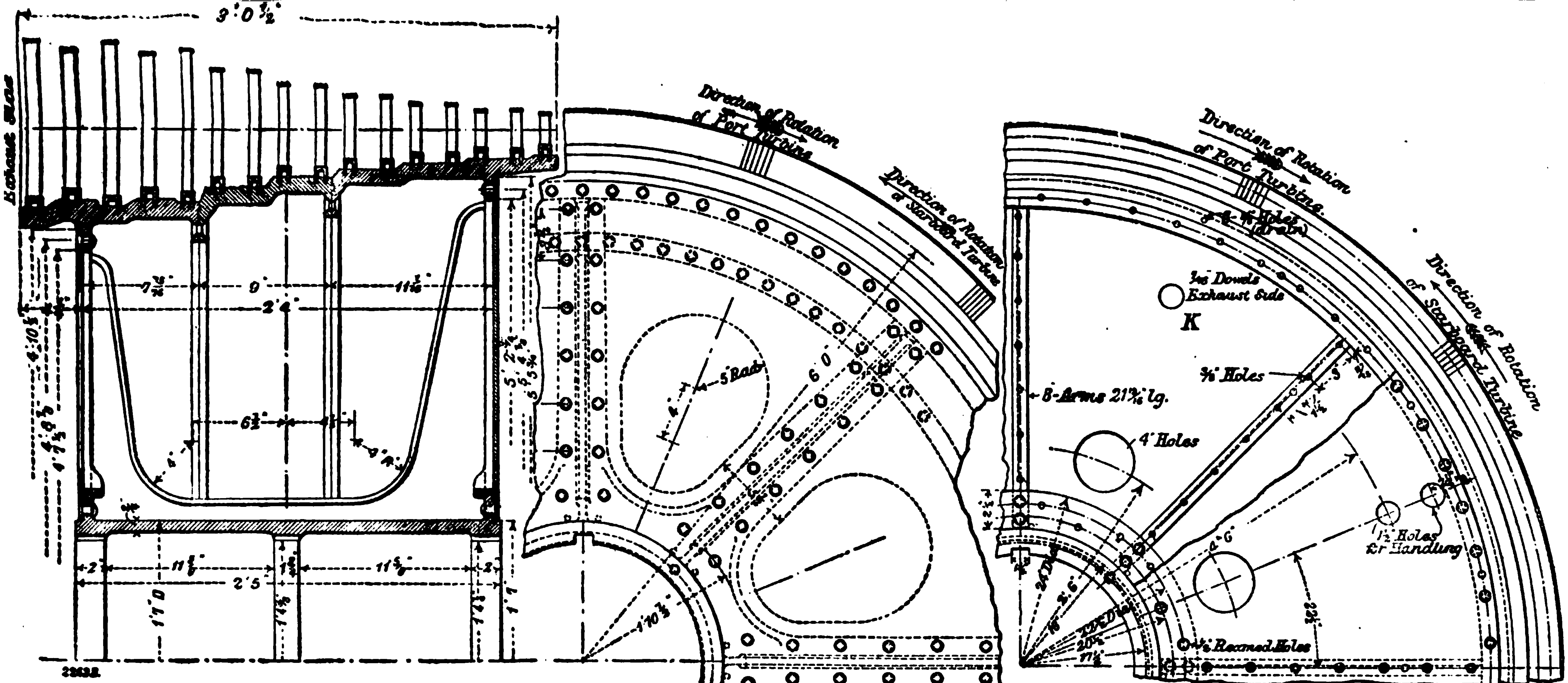
The drum is shrunk or forced on the shaft, and secured by suitable keys in the same fashion as the wheels.

The relative position of the drum and the various wheels is maintained by means of distant rings, and the whole, after being assembled, is kept in place by a large locking ring forward.

Nozzles and Blades.

The first stage nozzles, in which only partial admission takes place, are bolted to ribs formed on the casing; the ribs have suitable openings with easy fillets for the free flow of the steam to the nozzles. The nozzles of the full admission stages are bolted to the cylinder, and also carry the guide blades, the clearance between the nozzle plates and the rotor drum being kept down to a working minimum, to reduce leakage between the two sections.

Note.—In the majority of Brown-Curtis turbines at present under construction, the drum stages are fitted throughout with reaction blading similar to that of the Parson's type, the blades being fixed by the "dovetail" or undercut root method (pp. 446-447) in every case.



No. 82.—Drum Stage.*

(U.S. Destroyers, "Perkins" and "Sterrett.")

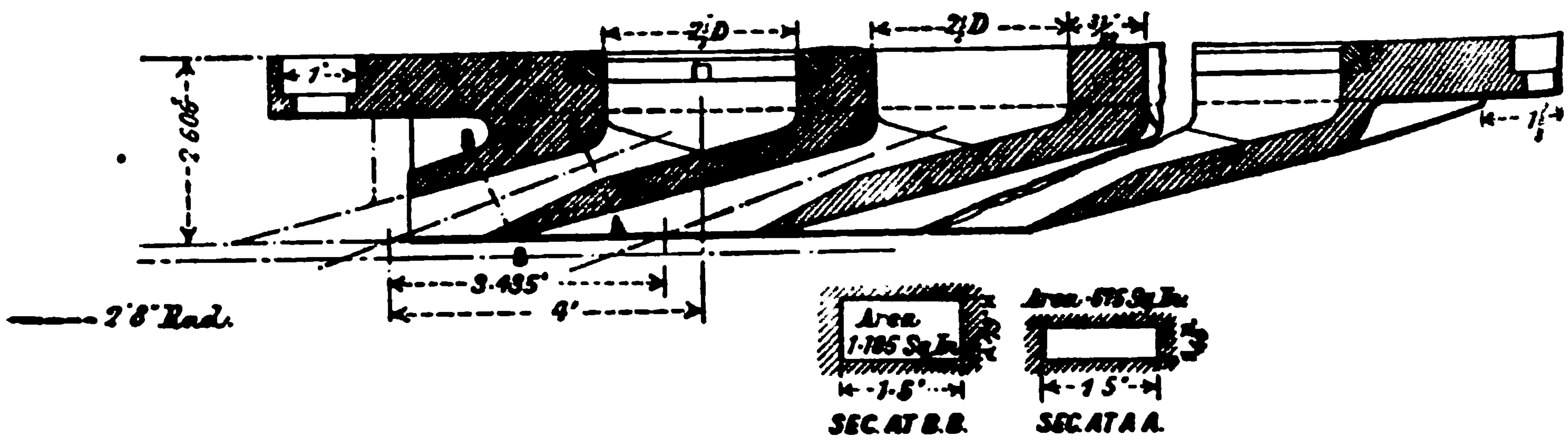
The drum section (Brown-Curtis) consists of 8 stages, as shown, each of 2 moving rows and 1 fixed row of blades. Notice that the drum and casing are both stepped, also that the blades are stepped. The forward end of the drum is solid, forming a disc, on the surface of which the steam pressure acts aft to counterbalance the propeller thrust forward. This arrangement is known as the Brown-Curtis design of impulse turbine.

Note.—In recent designs reaction blading has been substituted for nozzles and impulse blading as formerly fitted.

* Reprinted from *Engineering*.

Astern Stages.

The astern turbine, placed aft of the ahead, usually consists of two wheel stages, each with four or five moving blade rows and three or four fixed blade rows fitted in the same cylinder as the ahead turbine, and (similar to Parsons' practice) the exhaust to the condenser being common to both. The discs and diaphragms for the astern are similar to those for ahead, and are secured in position in the same manner.



No. 83.—Astern Turbine Nozzles.

(U.S. Destroyers, "Perkins" and "Sterrett.")

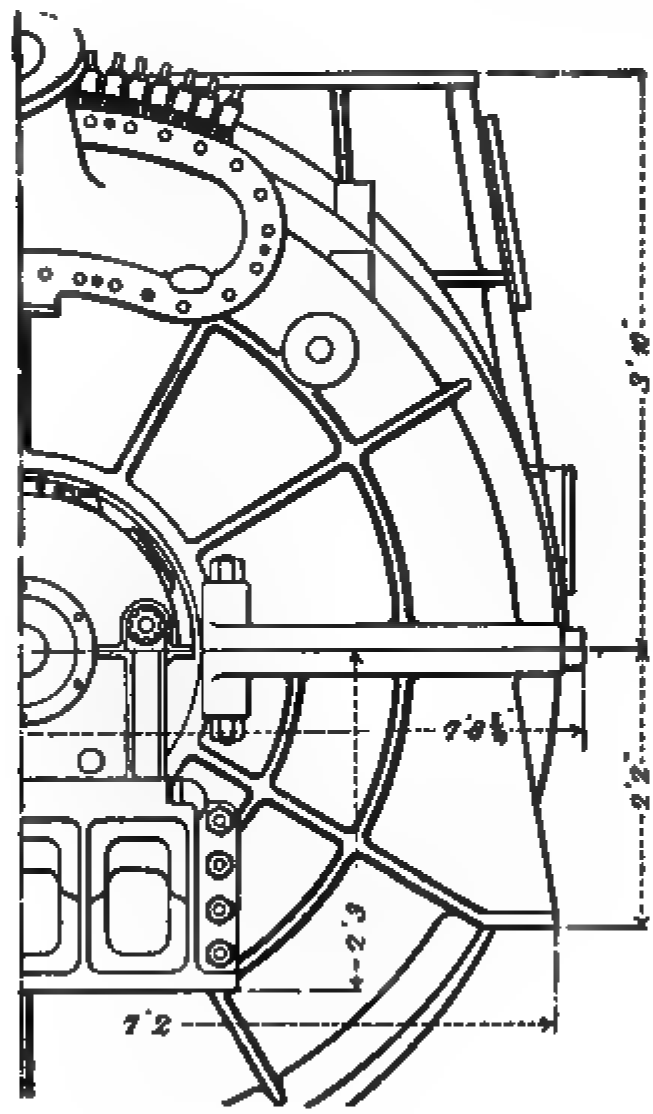
Expansion range of nozzle $= \frac{1 \cdot 185}{\cdot 675} = 1 \cdot 75$.

Nozzle Valves.

At the ahead end each nozzle is fitted with a hand-controlled flat shut-off valve, the number of nozzles and valves varying in number from fifteen to twenty-five. It will therefore be evident that even at low powers full pressure is admitted through each separate nozzle, no matter how few the number open may be. For low speeds and powers certain specially shaped nozzles are employed, called "cruising nozzles," to meet the altered conditions then obtaining. At the astern end the larger proportion of the nozzle openings have no valves, the remainder only being controlled by flat valves similar to those of the ahead, so that, when steam is supplied for astern running, the bulk of the nozzles are open to receive it without further manipulation being necessary.

Glands.

The turbine shafts are packed against steam leakage out, or air leakage in, by means of carbon block rings held in position by light springs. The carbon segments are also secured in position by keys as shown in the sketch. A steam "seal" is fitted between the carbon blocks working at a pressure of 3 or 4 lbs. above the atmosphere.



[*To face page* 438.

. 85.—Nozzle Control Slide Valves of Curtis Turbine (worked by hand.)

This drawing shows clearly the small hand control valves employed for these nozzle openings, also the nozzle and the initial steam admission branch to the nozzle box.

No. 86.—Carbon Block Packing Gland of Curtis Turbine.

1. Carbon segments.
2. Springs holding carbon segments.
3. Gland casing.
4. Flange cover.
5. Turbine casing.
6. Steam seal.

Drains.

Drains are led from the glands to one of the low-pressure stages, and each stage is drained to the following one; when the pressure is lower, the whole is finally led away to the condenser. These drains are only intended to be opened at intervals, and require to be kept tight shut when running, otherwise steam leakage between the stages would take place.

Thrust Block.

A small thrust block of the usual marine type is placed on the shaft forward of the turbine, but, as the propeller thrust is usually more than balanced by the steam pressure on the forward end of the drum stage, whatever pressure is transmitted to the thrust (unless at low powers) is taken by the *forward* side of the shoes in place of the after side. Therefore, as in the case of the Parsons turbine, the block is subject to small loads, so that a block of small bearing surface is sufficient to meet the conditions maintaining.

Clearances.

At the thrust block the longitudinal clearance is about $\frac{1}{16}$ in., but aft at the low-pressure end the clearance is more, being in some cases as much as $\frac{1}{2}$ in. This is to allow for the unequal expansion of the rotor and casing, which is arranged in this way to work aft.

The radial or blade tip clearance between casing and rotor, and between rotor and casing when measured cold, ranges from $\frac{50}{1000}$ in. at the first stages to $\frac{75}{1000}$ in. at the latter stages.

In the Curtis turbine the longitudinal clearance is of more importance than the radial clearance, owing to the fact that the pressure being nearly equal throughout a stage the leakage due to tip clearance is negligible.

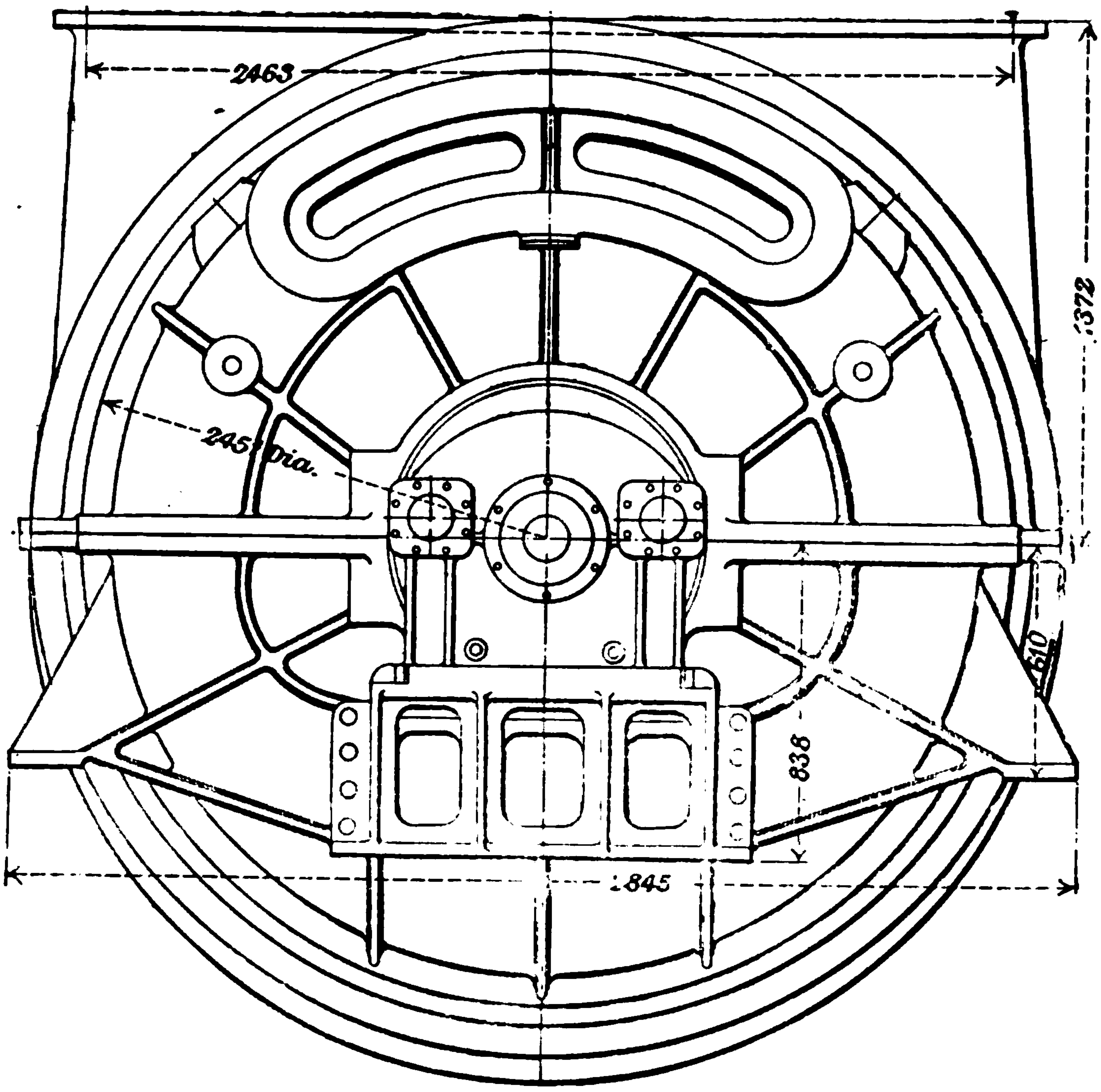
Speed Regulation.

The revolutions and speed are regulated by the number of nozzles open, and by means of a throttle valve of the usual marine construction. All nozzles in use should be open full, and those not in use shut tight.

The turbines of the U.S. battleship, "North Dakota," are fitted with twenty ahead nozzles and sixteen astern nozzles.

Of the ahead, four nozzles are open constantly, and the other sixteen are controlled by hand valves. Of the astern, ten nozzles are open constantly, and the other six are controlled by valves.

In reversing the ahead throttle valve is shut and the astern throttle valve opened, and as the blade construction is of great strength, no damage would result if by mischance both throttles



End View.

-Bixio."

mm. to 1 in

[To face page 440.

on five separate wheels, whilst all the moving blades of the remaining eleven pressure stages are fitted to a drum. The object of this arrangement is to obtain between the two ends of the drum a pressure drop to partially balance the propellor thrust and to reduce the load on the thrust-block. The two stages of the astern turbine consist of two wheels separated by a diaphragm. The steam inlet is at the

[1] Reprinted from *Engineering*.

top of the forward end of the ahead turbine, and at the top of the aft end of the astern one.

The casing is of cast iron, divided longitudinally into a lower and an upper half, each in three parts, as shown in No. 87. The ends are cast in one piece with the main casing, and are bolted to the supports for the bearings of the rotor shaft. The rotor consists of a hollow shaft of forged steel, on which are keyed the five wheels and the drum of the ahead turbine, and the two wheels of the astern turbine. There are at the forward end five thrust-collars. The drum is a steel casting having eight arms, fitted with a forged steel rim in four rings; the rim is held to the drum by two plates riveted on to the arms and rim. The plate fitted at the extreme forward end is entire, whilst the one at the aft end has eight large openings cut in it. It follows that when running ahead the front end of the drum is under the pressure which obtains in the sixth pressure stage, and the aft end under the pressure in the condenser, the turbine shaft being by this means under an end thrust, which opposes the propeller thrust. At all powers above half power the thrust of the steam exceeds the propeller thrust, and the residual thrust is taken up by the front surfaces of the four thrust-rings in front of the turbine.

The five diaphragms separating the first six pressure stages of the ahead turbine, and that which separates the two stages of the astern turbine, extend down to the shaft, with which they make

TABLE I.—CURTIS TURBINES OF THE

	Wheels.						
	1.	2.	3.	4.	5.	6.	7.
Absolute pressure in front of nozzles (lbs. per sq. in.)	273	94	67	48	33	23	15·8
Corresponding dryness fraction	0·98	0·9705	0·963	0·956	0·949	0·942	0·935
Absolute pressure at discharge (lbs. per sq. in.)	94	67	48	33	23	15·8	12·6
Velocity of steam at discharge (ft. per sec.)	2070	1132	1132	1132	1132	1132	922
Total section at throat of nozzles (sq. in.)	...	21·4	29·6	41	61	81	122
Diaphragm clearance (in.)	...	0·0149	0·0196	0·0248	0·0299	0·0299	0·0326
Net section available for leakage (sq. in.)	...	0·920	1·240	1·580	1·830	1·890	8·000
Mean diameter of blading	6 ft. 8 in.	6 ft. 8 in.	6 ft. 8 in.	6 ft. 8 in.	6 ft. 8 in.	6 ft. 8 in.	6 ft. 8 in.
Nozzle angle of discharge (deg.)	20	20	20	20	20	20	20
Radial height of nozzles (in.)	1·496	1·496	1·496	1·496	1·496	1·259	1·692
Axial width of blades (in.)	1 & 0·748	0·748	0·748	0·748	0·748	0·748	0·748
Rows of moving blades per stage	4	3	3	3	3	3	2
Arc covered by nozzles (deg.)	...	69	94	130	197	326	360

practically a steam-tight joint. They are made of sheet steel, dished and fitted, as shown. The ten diaphragms which separate the last ten stages of the ahead turbine extend down to the drum only. The steam thus has a leakage path of larger section than in the case of the five preceding diaphragms, but, on the other hand, the pressure of the steam has only about one-tenth of the value.

The forward end of the casing has an inlet branch cast in one piece with it, which contains the nozzles for the first ahead running stage, twenty in number, each nozzle being provided with a shut-off valve. By this means it is possible to reduce the number of nozzles opened, so as to maintain at low speeds the maximum steam pressure in the inlet branch, and thus prevent any loss due to throttling. The nozzles are made of Monel metal (an alloy of nickel and copper) screwed to a brass plate, which serves as a port face for the closing valves. Of the twenty nozzles, four are of a special type, and only come into operation at cruising speeds, for which, as is well known, the conditions of pressure in the various stages are entirely different from those which obtain at the high speeds. Of the remaining sixteen, it is only necessary to utilise twelve for the development of the normal maximum power of 7500 shaft-horse power at 450 revolutions. The other four are provided in case it is required to overload the turbine, utilising the maximum quantity of steam which can be produced with oil fuel combustion added to the ordinary coal

Scouts "Marsala" and "Nino Bixio."

Drum.								
8.	9.	10.	11.	12.	13.	14.	15.	16.
12·6 0·929	9·8 0·9235	7·6 0·918	6 0·912	4·6 0·906	3·6 0·901	2·8 0·8945	2·2 0·889	1·6 0·8835
9·8	7·6	6	4·6	3·6	2·8	2·2	1·6	1·2
922	922	922	922	922	922	922	922	922
154·5 0·0326	195·5 0·0326	248 0·0326	317·5 0·0326	406 0·0326	533 0·0326	680 0·0326	868 0·0326	1073 0·0326
7·750 ft. 8 in.	7·680 6 ft. 8 in.	7·640 6 ft. 8 in.	7·550 6 ft. 8½ in.	7·450 6 ft. 8¾ in.	7·300 6 ft. 9¼ in.	7·140 6 ft. 9½ in.	7·020 6 ft. 10 in.	6·990 6 ft. 11 in.
20 2·126 0·748	20 2·716 0·748	20 3·464 0·748	20 4·370 0·748	20 5·551 0·748	20 7·244 0·748	20 9·212 1	23 10·275 1	25 11·535 1
2 360	2 360	2 360	2 360	2 360	2 360	2 360	2 360	2 360

combustion. All the nozzles are rectangular in section, and of the convergent-divergent type, since the pressure ratio is less than the critical; from an absolute pressure of 273 lb per sq. in., the steam expands down to an absolute pressure of 94 lb. per sq in. in the nozzles of the ordinary type, and down to a pressure approximating to the atmospheric in the cruising speed nozzles. The total area of the 16 ordinary nozzles at the throat is 10 sq. in. whilst their full section is 1·145 times greater. The axis of the nozzle makes an angle of 20° with the plane of the first wheel.

Table I., gives a few figures concerning the details of the construction of the turbines for the "Marsala" and "Nino Bixio"; whilst in Table I. will be found data on the Curtis turbines for three types of ships—battleships, scouts, and destroyers.

TABLE II.—PRINCIPAL DIMENSIONS AND WEIGHTS OF MARINE CURTIS TURBINES.

	Battleships.		Scouts.	Destroyers.
	"North Dakota."	"Rivadavia" and "Moreno."	"Marsala" and "Nino Bixio."	"Perkins," "Sterrett," and "Walke."
Number of shafts	2	3	2	2
Designed full power total S. H. P.	25000	45000	22500	12000
Over-all length of turbine	31 ft. 9 in.	30 ft. 1 in.	21 ft. 9 in.	19 ft. 8 in.
Distance between centres of main bearings	22 ft. 6 in.	22 ft.	16 ft.	15 ft.
Maximum width	17 ft. 1 in.	15 ft. 4 in.	9 ft. 4 in.	7 ft. 9 in.
Maximum height	14 ft. 11 in.	13 ft. 11 in.	9 ft. 2 in.	7 ft. 8 in.
Mean diameter of blading	12 ft.	11 ft.	6 ft. 8 in.	6 ft.
R.P.M. at full power	245	275	450	610
Weight of one turbine complete	tns. c. q. 178 12 3	tns. c. q. 177 3 1	tns. c. 46 15	tns. c. 30 7
Weight per S.H.P. at designed full power	31·95 lbs.	26·45 lbs.	13·95 lbs.	11·35 lbs.

Parsons' Combined Impulse and Reaction Turbines (High Speed Craft).

The combining of impulse and reaction blading in the same turbine represents the most recent development of the Parsons' system, and the chief characteristics of this type may be summarised as follows:—

1. The total expansion of steam from the H.P. initial pressure to the condenser pressure is carried out in one turbine, in place of in two or more turbines as formerly.

2. The complete installation consists of two independent turbine casings, two shafts, and two propellers. Each turbine casing contains one ahead turbine forward, and one astern turbine aft (4 turbines in all).

S.

5.

No. 88B.—Parson's Combined Impulse and Reaction Turbines.

End view of turbine showing nozzle control valves, initial steam strainer (centre), and independent steam-pipe connections to each control valve, also exhaust bend and condenser.

No. 88C.—Parson's Combined Impulse and Reaction Turbines.

Observe astern impulse wheel on left, followed by a few reaction expansions; also the ahead impulse wheel on right, followed by reaction expansions, usually eight or nine in number.

3. Each ahead and astern turbine consists of an impulse wheel stage followed by a drum stage with the usual reaction blading.

4. The ahead and astern impulse stage usually consists of 4 moving blade rows and 3 fixed blade rows, or 7 rows in all.

5. The drum stage bladings consists of from 8 to 9 expansions, each formed of 8 or 9 blade rows at the initial stages, and falling off to perhaps 7 rows at the 7th expansion, and 3 or 4 rows at the 8th and 9th expansion.

6. The first five expansions have blades of increasing height, the ratio being about 1 : 1·5.

7. The last three or four expansions have blades of equal height to those of the 5th expansion, but the openings for steam flow are increased by the fitting of "semi-wing" and "wing" blades, the blade row pitch being also greater.

8. If the initial steam is, say, 220 lbs., the pressure at nozzle discharge will be, under full power conditions, about 80 lbs.; and this, it should be noted, is also the pressure at the end of the impulse stage, forming at the same time the initial steam for the first expansion of the drum stage. This pressure is termed "steam at 1st stage."

9. The steam speed at admission to impulse stage is about 1800 ft. per sec., and at admission to first reaction expansion stage about 280 ft. per sec.

10. At full power the blade speed at the impulse stage is about 200 ft. per sec., and at the 1st expansion of the reaction stage about 160 ft. per sec.

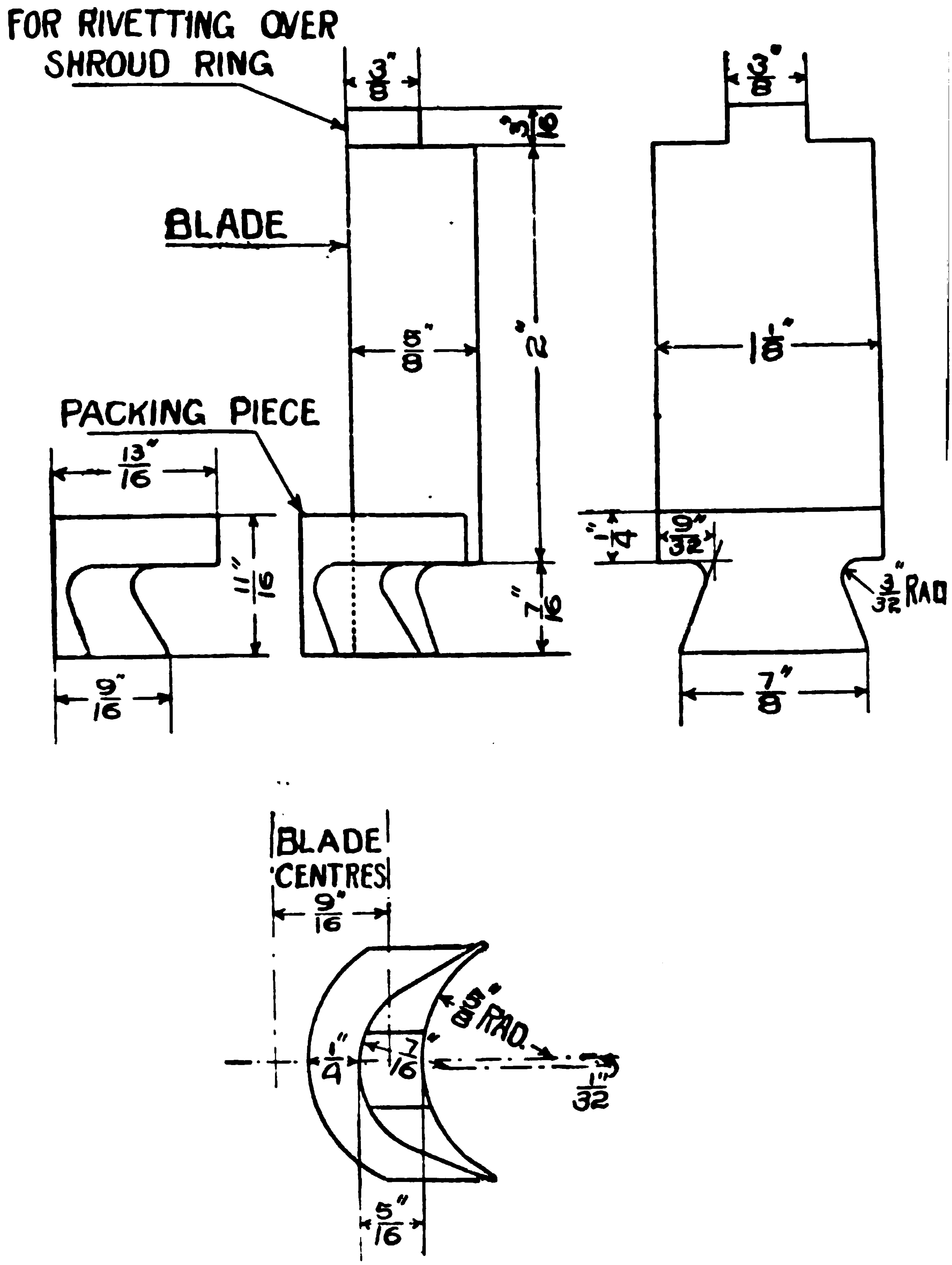
11. The increase in blade height at the impulse wheel stage is about $\frac{1}{2}$ in. per blade in succession, except at the last blade row (moving), which is only $\frac{3}{16}$ in. more in height than the previous fixed blade rows: this characteristic also maintains in the case of the Curtis turbine blading.

12. As with the Curtis system, control valves are fitted on the nozzle box at the top half of the casing, by means of which the quantity of steam at constant pressure entering the turbine can be regulated according to the power and speed required.

13. The astern turbine consists of a set of impulse blading similar to the ahead, but often of smaller mean diameter across blades, and of four expansions of the usual reaction blading on the drum stage, the first three expansions being of increasing blade height, and the last two (3rd and 4th) of equal blade height but of "semi-wing" or "wing" type.

14. The blade tip clearance at the impulse stage is kept at about $\frac{3}{32}$ in. or $\frac{93}{1000}$ in., and at the reaction stages may vary from about $\frac{60}{1000}$ in. (·06 in.) at the first expansion to $\frac{90}{1000}$ in. (·09 in.) at the last expansion.

15. The ahead "dummy" rings are placed forward of the impulse stage nozzles.



No. 90.—Curtis Type Blading.

For Circumferential Caulking.

(Full size.)

1. Groove in rim of impulse wheel.
2. Opening for insertion of blade packing pieces.
3. Blade (with undercut root).
4. Packer.
5. Top edge of packer.

The types of blades shown are fixed by means of "circumferential" caulking in place of "radial" caulking, and the enlarged opening (2) is closed up after the blades and packers are all assembled by a closing key or caulking piece which is hammered down and locked in place.

Note.—The blades can be entered in the grooves by turning them sideways, but the packers can only be entered by means of the specially cut opening (2).

Open and Closed Exhausts (48).

In naval practice, when developing full ahead power, the turbines are run under "closed exhaust" conditions, which means that the auxiliary engine exhausts (pumps, steering gear, etc.) are led into one of the turbine expansions, and the exhaust steam then does work in the turbine before finally passing to the condensers.

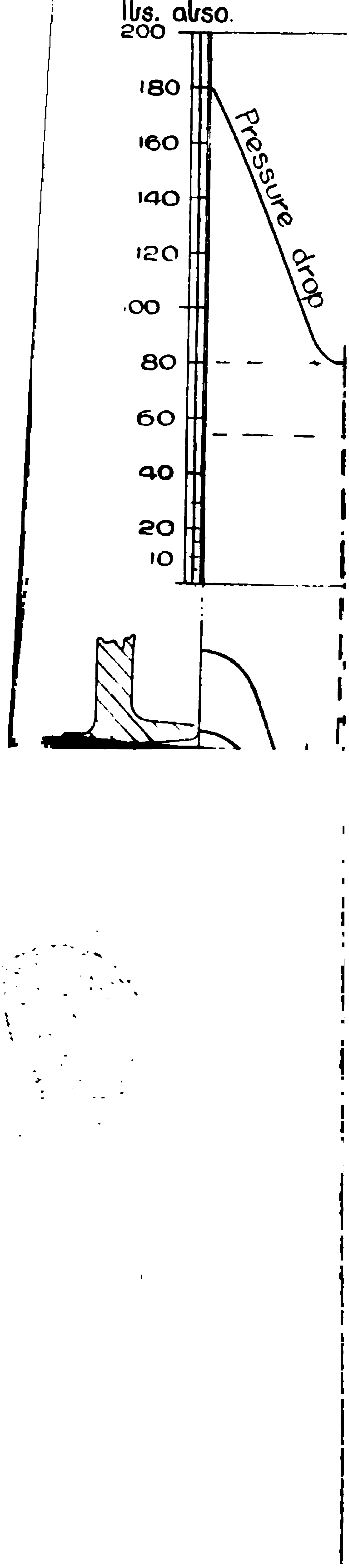
"Open exhausts" means that the auxiliary exhausts are led to the condensers direct.

With combined impulse and reaction turbines the exhaust steam usually enters the turbine at the 3rd expansion, an exhaust belt being provided for that purpose as shown in sketch.

The difference in pressure resulting from working with open and closed exhausts are shown below :—

	H.P. Steam.	1st Expansion Steam.	3rd Expansion Steam.
Open exhausts -	200 lbs.	64 lbs.	8 lbs.
Closed " -	200 "	64 "	13 "

Note.—"1st expansion" means the pressure entering the *first* row of reaction blading, and which is equal to the pressure obtaining at the impulse stage.



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SECTION IX.

GEARED DOWN TURBINES.

To allow of combined high turbine speeds and low propeller shaft speeds geared down turbines have recently been introduced. This arrangement admits of economy at low ship speeds, owing to the fact that turbines are most efficient at high revolution speeds, and propellers most efficient at low revolution speeds.

Arrangement.—In the single propeller shaft arrangement one side H.P. ahead turbine and one side L.P. ahead turbine are connected by two small gear wheels to two large gear wheels secured to the centre driving shaft. The turbines run at about 1200 revs. per min. and the propeller shaft at 60 revs. per min., the gear-down ratio thus being as 20 : 1, because $1200 \div 60 = 20$. In the two propeller shaft arrangement the above system is usually duplicated, two H.P. and two L.P. turbines being fitted and connected up similarly. The turbines and gear wheels are joined up by flexible type couplings. The gear wheels are enclosed in a casing, and an oil service under pressure is sprayed in jets on to the contact surfaces of the wheels. Thrust blocks are fitted near the forward end of the propeller shaft, to take up the thrust, and the turbines are balanced by steam pressure acting on differential type dummy pistons. The helical-toothed gear wheels are very accurately cut out of hard steel by a machine specially designed for the purpose. The astern turbines are arranged as in ordinary turbine practice, being inside the ahead turbine casing, and fitted with separate steam connections, etc.

It may be pointed out that the geared down arrangement can be adapted for either low or high ship speeds, but is not so necessary at high speeds as it is at low speeds. The gearing down allows of speeds of 10 or 12 knots with reasonable economy, whereas at these speeds and direct turbine drives the economy would fall off for the reasons mentioned previously.

Gear Wheel Teeth.

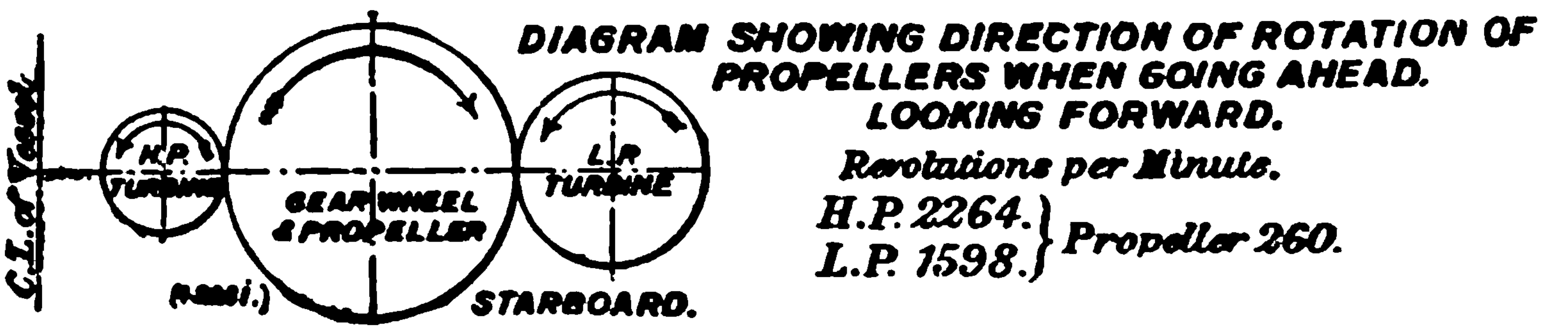
Parsons' gear wheel teeth are of helical formation, the steps of which merge into a spiral, and, as the contact points are constantly

changing, the contact is of a rolling nature, no sliding action whatever taking place: friction is therefore reduced to a minimum.

As, however, the teeth lie at an angle, end thrust is set up, but this is corrected by fitting double sets of teeth and pinions with the angle of teeth opposed to each other, so that the end thrust is balanced automatically. In any pair of wheels the angle of helix formed by the teeth must be the same in both pinions, even though the gear ratio on each side be different.

The large gear wheel is constructed of carbon steel of 30 to 35 tons tensile strength, and possessing an elongation of 26 per cent.

The pinions are constructed of nickel steel of 40 tons tensile strength, an elastic limit of 24 tons, and possessing an elongation of 25 per cent.



No. 93—Gear Wheels of T.S.S. "Ciudad de Buenos Aires."

Gear-down ratio H.P., 8·7 to 1. Gear-down ratio L.P., 6·1 to 1.
H.P. turbine revs. = 2264, and 2264 ÷ 8·7 = 260 shaft revs.
L.P. " = 1598, " 1598 ÷ 6·1 = 260 "

In the "Tuscania" the gearing is of the double helical pattern, the teeth having a spiral angle of about 44°, depth of teeth engaged $\frac{3}{8}$ in., and the peripheral speed is about 3700 ft. per minute.

Efficiency.

The mechanical efficiency of the Parsons' gear wheels is stated to be fully 98 per cent.

Racing.

The inertia effect of the large gear wheel acts somewhat in the manner of a flywheel and tends to reduce the tendency to "race" when the propeller rises out of the water. The weight of the rotor drums also act similarly.

The following extract descriptive of the gear hobbing machine is reprinted from a paper by the Hon. Sir Charles Parsons, K.C.B., and read before the Institution of Naval Architects, 13th March 1913.

"An improved method of cutting gear-wheel teeth has, however, been developed by the author and his colleagues, which must now be described. Primarily, it aimed at destroying the periodicity of the errors, but incidentally

94.—Sectional Elevation, also Plan of Gear Wheel Teeth-Cutting Machine.

it also accomplishes a considerable reduction of the errors themselves present in the parent gear.

"It will be seen that in the process ordinarily adopted, in which the work is mounted on a table rotated by means of a worm and worm-wheel, the latter being attached permanently to the table, the errors will be some function of the angular position of the work, and therefore lie in planes through the axis of rotation; and if, as is mostly the case, the errors of the parent gear are periodic, these planes will lie at equal angular intervals, and will come into mesh periodically. Now it will be seen that, if the work is given a small steady advance in relation to the table, the errors, instead of lying in planes through the axis, will lie in spirals around the wheel, and that when put to work they will be obliterated, and leave a true wheel.

"No. 94 shows illustrations of the adaptation of this new principle of cutting to an existing gear hobbing machine. A secondary table is mounted on the original table of the machine, and given a creep in advance of 1 per cent. in relation to it by means of the train of gearing shown, the main worm driving the lower table being driven at 1 per cent. less speed, so as to secure the same rotational speed as before the creep was introduced. While the most important effect of this arrangement is that the errors in the teeth will lie in very oblique spirals around the wheel, resulting in great uniformity in the gearing, at the same time it has also an important effect in reducing the errors themselves.

"If the periodic error in the worm-gear of the original table be represented by a sine curve with a period corresponding to the teeth of the worm-wheel, that is 160 per revolution, an advance of 1 per cent. results in the phase of the error being shifted 1·616 of a complete pitch at each revolution of the work. With the cutter advancing across the wheel, the result is a series of overlapping cuts of varying depth, the maximum depth being, say, about $\frac{4}{1000}$ in. below the minimum. These have been represented on the diagram, the advance of the cutter across the wheel being taken at $\frac{1}{20}$ in. per revolution, whilst the amplitude of the error, and therefore the position of the cutter, is represented greatly enlarged in the vertical direction. It will be seen from this how the lowest positions predominate, and a series of cuspidal ridges remains of about one-fifth the magnitude of the original errors."

Report on Noise of Gear Wheels.

"At anything up to 16 knots the gear is practically noiseless, the counter gear having to be referred to to see if turbines are running."

"At 16 to 20 knots there is a slight hum from the gear wheels, and at 20 to 23 knots there is a distinct, although not considerable, buz noticeable. On the whole, the gear is remarkably quiet."

The above report affords good testimony to the mathematical accuracy of the gear wheel teeth cutting.

Economical Speed.

The most economical ship speed for geared down turbines is understood to be about 15 knots.

… was built in 1887 by Messrs Short Brothers, of Sunderland. Her dimensions are:—Length on load water line, 275 ft.; breadth, moulded, 38 ft. 9 in.; depth, moulded, 21 ft. 2 in.; mean loaded draught, 19 ft. 8 in.; and displacement, 4350 tons. The boilers—two in number—are 13 ft. diameter by 10 ft. 6 in long, with a total heating surface of 3430 sq. ft., and grate area of 98 sq. ft., working under a pressure of 150 lbs. with natural draught. The

The above report affords good testimony to the mathematical accuracy of the gear wheel teeth cutting.

Economical Speed.

The most economical ship speed for geared down turbines is understood to be about 15 knots.

Flexible Coupling (96).

With geared down turbines the turbine spindles and gear pinion shafts are usually coupled together by means of a flexible coupling of the Parsons' type, as shown in the sketch. The coupling illustrated consists of four separate parts, two for either side, and is of the "star" or "claw" pattern, the teeth of the outer portion engaging with those of the inner portion, and so transmitting the torsional stress from one shaft to the other. It will be noticed that the teeth of the inner keyed-on part, project out radially, and fit in between the dovetail-shaped teeth of the outer flanged part of the coupling. Oil holes are provided for lubrication, and outer baffle plates cover the ends of the coupling. This coupling arrangement allows of longitudinal change of position of the rotor, due to either heat expansion or unequal balance of steam pressure, and also allows of dummy adjustment.

No. 96.—Claw Type Flexible Coupling.

1. Radial claws.
2. Circumferential claws.

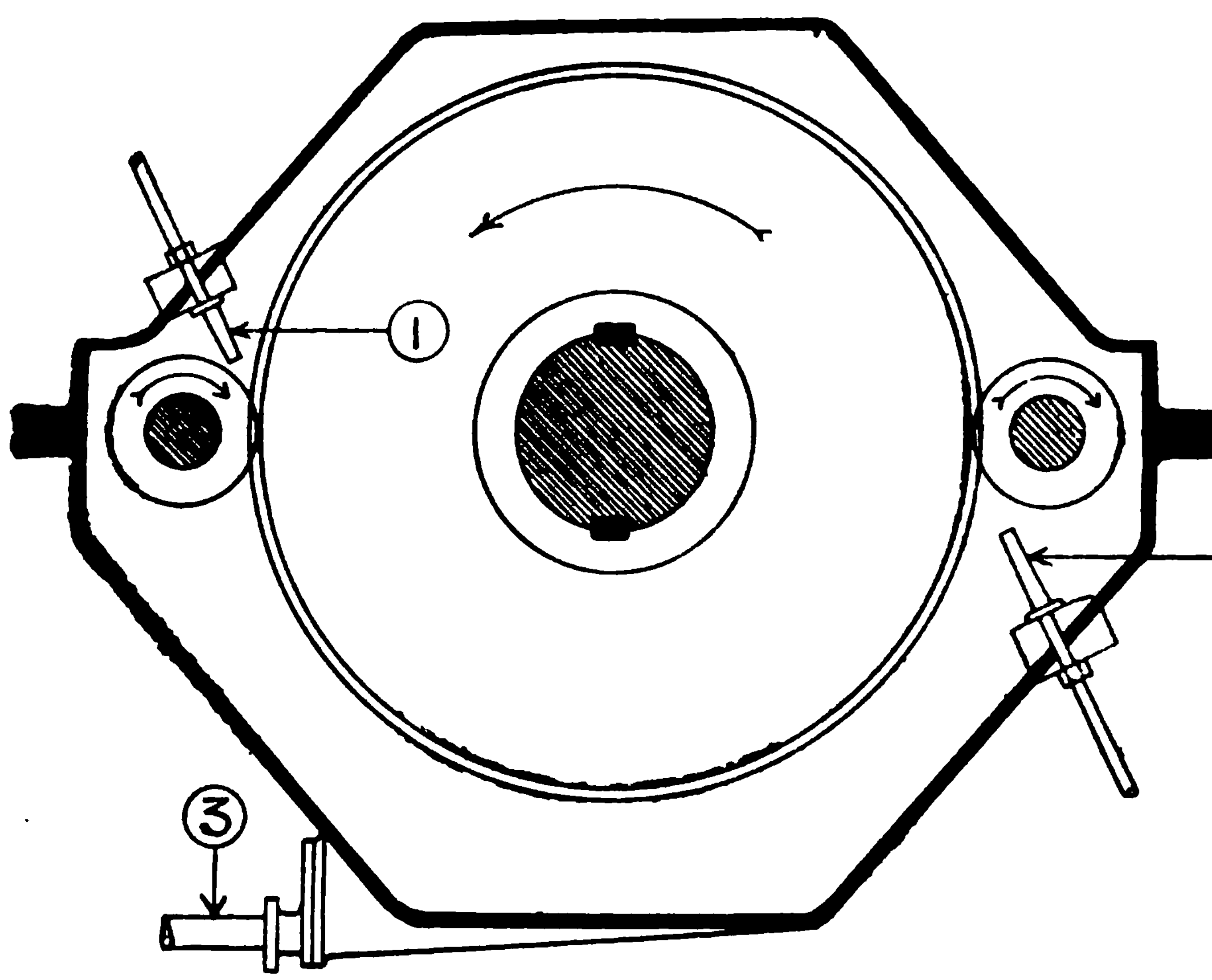
Geared Turbines, S.S. "Vespasian" (98).

The "Vespasian" was built in 1887 by Messrs Short Brothers, of Sunderland. Her dimensions are:—Length on load water line, 275 ft.; breadth, moulded, 38 ft. 9 in.; depth, moulded, 21 ft. 2 in.; mean loaded draught, 19 ft. 8 in; and displacement, 4350 tons. The boilers—two in number—are 13 ft. diameter by 10 ft. 6 in long, with a total heating surface of 3430 sq. ft., and grate area of 98 sq. ft., working under a pressure of 150 lbs. with natural draught. The

propeller is of cast iron, and has four blades, having a diameter
14 ft., pitch 16·35 ft., and expanded area of 70 sq. ft.

The propelling machinery consists of two turbines in "seri
viz., one high-pressure and one low-pressure, the high-pressu
turbine being placed on the starboard side of the vessel and the lo
pressure on the port side. At the after end of each of the turbi
a driving pinion is connected, with a flexible coupling between
pinion shaft and the turbine, the pinion on each side of the ves
being geared into a wheel, which is coupled to the propeller shaft.

A reversing turbine is incorporated in the exhaust casing of
low-pressure turbine. The air, circulating, feed, and bilge pu



No. 97.—Gear Wheel Case.

1. Pressure oil jets.
2. " "
3. Oil drain.

are of the usual design for tramp steamers, and are driven by m
of a crank and connecting rod coupled to the forward end of g
wheel shaft. The turbine and pinion shaft bearings are under fo
lubrication, similar to ordinary turbine practice. The teeth of
pinions and of the gear wheel are lubricated by means of a "spra
pipe extending the full width of the face of the wheel. Independe
oil pumps are fitted for supplying oil to the bearings and gear whe
with a view to the possibility of experimenting with differe
lubricants for the gear wheel, the oiling system for the bearings bei
separate from that of the gear wheel.

ditions of the service demand ample astern power, and, in order to provide for these requirements, this has been arranged to be not less than 60 per cent. of the maximum ahead power.

The high-pressure turbine is 3 ft. maximum diameter by 13 ft. over all length, and the low-pressure 3 ft. 10 in. by 12 ft. 6 in. length. The turbines are similar in design to a land turbine, being balanced for steam thrust only, the propeller thrust being taken up by the ordinary thrust-block of the horse-shoe type which is fitted aft of the gear wheel.

The cooling surface of the condenser is 1165 sq. ft.

The gear wheel is of cast iron, with two forged steel rims shrunk on. The diameter of the wheel is 8 ft. 3½ in. pitch circle, having 398 teeth—double helical—with a circular pitch of ·7854 in. The total width of face of wheel is 24 in.; inclination of teeth 20° to the axis.

The pinion shafts are of chrome nickel steel, 5 in. diameter pitch circle, with 20 teeth ·7854 circular pitch The ratio gear is 19·9 to 1.

On the first four voyages careful measurements of water consumption were taken. The following table gives the data and results obtained on these voyages:—

Geared Turbines of S.S. "Vespasian."

RESULTS OF FIRST FOUR VOYAGES.

Date	9/6/10	16/6/10	16 6/10	22/6/10	22/6/10	22/6/10	29/6/10
Speed by log, knots	9·35	9·22	10·58	9·61	9·27	10·22	9·37
Revs. per minute	65·0	64·9	73·0	64·8	63·85	70·6	62·9
Boiler pressure, lbs. per square inch	137	135	145	135	135	140	135
High-pressure turbine (initial pressure, lbs. per square inch)	86	86	121	86	86	111	81
Vacuum (in inches)	28·5	29·1	28·6	28·55	28·4	28·4	28·3
Barometer	30·01	30·5	30·52	29·9	29·9	29·88	29·6
Water, main engines, lbs. per hour	12,140	12,300	15,680	11,890	11,730	14,510	11,100
Shaft horse-power	740	736	1,080	735	710	960	668
Water consumption, lbs. per S.H.P.	16·4	18·0	14·5	16·2	16·5	15·1	16·6

Geared Turbines of the "King Orry."

The machinery is on the twin-screw system, and includes two high-pressure and two low-pressure turbines of the latest Parsons combined impulse and reaction type. The arrangement of the turbines and gear is shown on the two-page plate. The two high-pressure turbines are in the centre, and the two low-pressure turbines, with which the astern turbines are incorporated, in the wings. The conditions of the service demand ample astern power, and, in order to provide for these requirements, this has been arranged to be not less than 60 per cent. of the maximum ahead power.

The high-pressure rotors are solid steel forgings, and bladed on two diameters, 14 in. and 23 in. Three expansions of eleven rows each are fitted on the smaller of these, and three expansions of four rows each on the larger. The mean diameter of the blading ranges from 1 ft. $3\frac{5}{16}$ in. on the high-pressure end to 2 ft. $2\frac{1}{4}$ in. at the other, in each turbine. As it is not possible to utilise the propeller thrust to balance the steam thrust, owing to the presence of the gear box and its fittings, the whole of the steam thrust in this type of turbine is designed to be taken on the dummies. As the rotor is solid, equalising pipes are led from the end of the third expansion to the forward side of the after dummy, and from the exhaust end to the forward side of the forward dummy, thus keeping the rotor in a practically floating condition axially. A small thrust bearing, as is usual, is fitted at the forward end of each turbine to take the pressure due to any variation in the speed. The arrangement is clearly shown in the illustration.

The low-pressure turbines have six expansions of reaction blading for going ahead, each having three rows of blades. The diameter of the drum is 3 ft. 1 in., and the mean diameter of the blading ranges from 3 ft. $3\frac{3}{16}$ in. to 3 ft. $7\frac{1}{4}$ in. The astern turbines have each three rows of impulse blading, followed by four expansions having four rows each of reaction blading. The mean diameter of the impulse wheel is 3 ft. 4 in., and in the case of the reaction blading the mean diameter ranges from 2 ft. $1\frac{1}{16}$ in. to 2 ft. $4\frac{1}{4}$ in. on a drum 2 ft. in diameter. The low-pressure rotors are of the more usual hollow type, and therefore do not need the equalising pipes as fitted in the case of the high-pressure rotors.

Each high-pressure turbine is designed to run at 2210 revs. per min., and each low-pressure at 1617 revs. per min.; each is connected up with its pinion shaft through a flexible coupling to correct any small want of perfect alignment. The gearing is of the usual type in two parts, with oppositely cut helices to neutralise end thrust, and the pinion-shaft bearings are arranged in this case to be of floating type with the object of equalising the pressures between the working teeth, and preventing objectionable noise. The pinion shafts, on which the teeth are cut from the solid, are of special nickel steel, and the wheels into which they gear are of forged steel. All these forgings were made at the Sheffield works of the builders, and the gearing was cut by the Parsons Company. The high-pressure pinions have 30 teeth, the low-pressure pinions 41, and the wheels 221, the arrangement thus providing for the essential hunting teeth, the ratios between turbines and propellers being—high pressure 13·58 to 1 and low pressure 5·39 to 1. The circular pitch of the teeth is 0·815 in., and the spiral angle $44^\circ\ 2\frac{1}{2}'$. Each pinion shaft is supported by three bearings of an aggregate length of 3 ft. $0\frac{1}{2}$ in. by $5\frac{1}{2}$ in. in diameter, the width between the teeth at each side of the centre bearing being $15\frac{1}{2}$ in. The usual thrust block for taking the thrust from the propeller is fitted immediately aft of the gear box,

The cruising turbines are geared down to the L.P. main turbine shafts by means of pinions and gear wheels placed forward. A clutch type coupling connects up the cruising gear wheel, which latter is disconnected (by means of a hand lever) when running at full power with the main turbines.

teeth is 0·815 in., and the spiral angle $44^\circ\ 2\frac{1}{2}'$. Each pinion shaft is supported by three bearings of an aggregate length of 3 ft. 0½ in. by 5½ in. in diameter, thc width between the teeth at each side of the centre bearing being 15½ in. The usual thrust block for taking the thrust from the propeller is fitted immediately aft of the gear box,

as may be seen from the engravings. The whole of the gear is enclosed in an oil-tight casing, and lubrication arranged for by sprays designed to maintain a film of oil continually between the working surfaces of the teeth. Messrs B. R. Vickers & Sons' patent frictionless stern glands are fitted to the propeller shafts to ensure that nothing in the way of preventable loss may take place here.

The condensers are of the Weir "Uniflux" type, having steel shells and cast-iron doors, and are guaranteed to maintain a vacuum of 28½ in. with inlet water at 60° Fahr. Two of Messrs Weir's dual air pumps are fitted, 24 in. in diameter by 15 in. stroke, and the same makers have supplied the main feed and forced-lubrication pumps. The circulating water for the condensers is supplied by two of Messrs Allen's 18-in. "Conqueror" centrifugal pumps, and they have also supplied the two 6-ft. diameter forced draught fans, and the dynamo and electric light engine. The vessel is fitted with two bilge and ballast pumps, a water service fresh-water pump, and a sanitary pump. An exhaust feed-water heater is fitted for utilising the exhaust from the auxiliary engines.

There are three boilers—two double-ended and one single-ended, with a total collective heating surface of 14385 sq. ft., and total great area of 383 sq. ft. The working pressure is 170 lbs., with a closed stokehold system of forced draught. The removal of the ashes from the stokehold is effected by two "Sentinel" ash hoists, which are fitted in the usual manner in the ventilators. A vertical auxiliary boiler, by Messrs Cochran, of Annan, is installed to run the electric light engine, etc., when the vessel is in port.

Very careful consideration has been given to the choice of speeds for turbines and propellers, and it is confidently expected that the arrangement will prove to be a highly efficient combination, showing considerable economy in working compared with direct-driven types working in similar vessels.

Torpedo Boat with Geared-Down Cruising Turbines.

One type of destroyer is fitted with three shafts, five ahead turbines, and two astern turbines.

The ahead turbines consist of—

- One H.P. main turbine (centre).
- Two L.P. main turbines (outboard).
- Two H.P. cruising turbines (outboard) and forward of L.P. turbines.

The cruising turbines are geared down to the L.P. main turbine shafts by means of pinions and gear wheels placed forward. A clutch type coupling connects up the cruising gear wheel, which latter is disconnected (by means of a hand lever) when running at full power with the main turbines.

1. At full powers (and gear wheels disconnected) the steam flow is as follows:—

The centre H.P. turbine exhausts to each wing L.P. turbine.
Each L.P. turbine exhausts to its own condenser.

2. When running under cruising conditions the steam flow is as follows:—

Each wing cruising turbine exhausts to the main centre H.P. turbine.
The centre H.P. turbine exhausts to each wing L.P. turbine.
Each L.P. turbine exhausts to its own condenser.

As previously mentioned, the gear wheels are disconnected from the main turbine shafts when running otherwise than cruising, although in cases of urgency, if steam is given to the main turbines direct without disconnection having first been made, the only result would be a slight reduction in shaft revolution speed due to the blade windage resistance of the cruising turbines when revolving at greatly increased speed. A non-return valve fitted between the H.P. and cruising turbines prevents steam flow back from the one to the other.

Impulse blading and control nozzles are fitted on the cruising turbines, on the H.P. turbines, and on the astern turbines, and reaction blading only on the L.P. turbines.

The revolution speed at full powers is about 450 per min., and the speed of vessel fully 33 knots. The gear-down ratio of the cruising turbines is as 12 : 1, so that at, say, a shaft revolution speed of 450 per min., the cruising turbine speed would be 5400 per min., as—

$$450 \times 12 = 5400.$$

As is customary with geared-down turbines, the cruising turbine is connected to the pinion wheel shaft by means of a sliding coupling which allows for adjustment and for expansion (see p. 453).

In another design of torpedo boat turbines in place of two H.P. cruising turbines, one H.P. and one I.P. cruising turbine are fitted, so that the steam flow under cruising conditions is as follows:—

Boiler steam to H.P. cruising turbine.
H.P. cruising turbine exhaust to I.P. cruising turbine.
I.P. cruising turbine exhaust to H.P. main turbine.
H.P. main turbine exhaust to each L.P. turbine.
L.P. exhaust to condenser.

In torpedo boat practice a separate cruising turbine is sometimes fitted for economical running at low speeds, and this turbine is geared down to one of the two shafts by means of a pinion engaging with the gear wheel at a different position from the pinions of the H.P. and L.P. turbines.

A clutch coupling connects up the cruising turbine spindle and the pinion shaft, and this is, of course, disconnected when running ahead at full power with the main turbines.

The cruising turbine is placed above the main turbines, and the pinion gears with the large shaft wheel on the upper part of the periphery, as shown in the sketch.

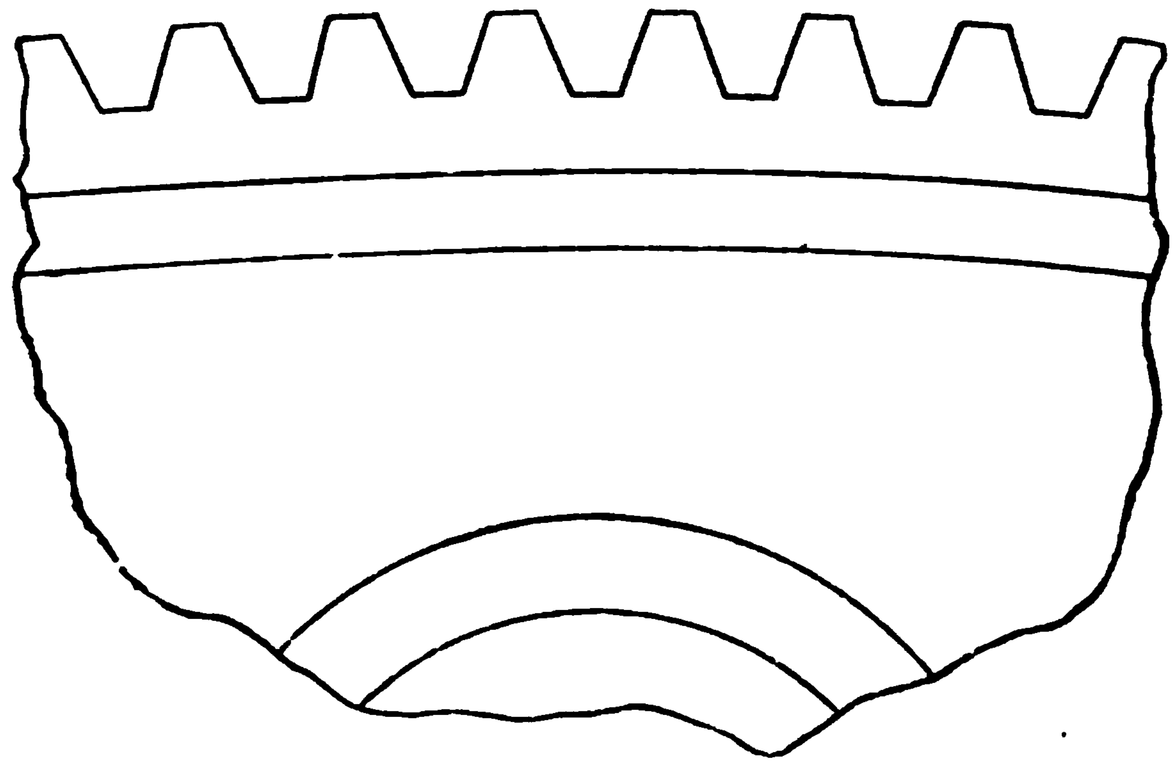
Turbine).

1. H.P. turbine steam.
2. H.P. exhaust to L.P. turbine.
3. Cruising turbine (for low speeds).
4. Cruising steam (from boilers),
5. Cruising exhaust to H.P. turbine.
6. Gear wheel and pinions.
7. Cruising turbine pinion.
8. Sliding coupling for throwing cruising turbine out of gear from pinion.

Geared Turbines of "Cairnross."

The main machinery of the "Cairnross" consists of two reaction turbines in series. These are arranged one on each side of main line of shafting, and each is connected by a flexible and sliding coupling to a pinion shaft; each pinion shaft is carried on three bearings, and between these bearings and pinion teeth are cut on enlarged parts of the shaft. The teeth are spirally cut, and as the spiral on forward pinion on each side is cut "to opposite hand" as compared with that on the after pinion, each pair forms virtually a "double helical" set, with the apex of the spirals cut away. These pinion shafts are of nickel steel. The large wheel driven by these pinions is on a shaft forward of, and coupled by usual flange coupling

and bolts to, the thrust shaft. This wheel consists of a heavy cast iron boss and pair of discs, on to which a mild cast steel hoop is shrunk and pegged. In this hoop the gear teeth are cut. The pinion and wheel-shaft bearings are incorporated in a heavy cast iron gear case, which gives them suitably rigid support and encloses the gears so that they may be run under jets of oil and kept free from risk of damage either by dirt or by the accidental introduction of any object that might endanger the gearing. The form of the teeth is involute. This form of tooth has one great advantage for machine cut gearing, namely, that the correct worm to work with an involute worm wheel, being geometrically equivalent to a "rack," has straight sides to its thread. A simply produced, multiple-threaded, hobbing cutter, of "worm" type, can therefore be used



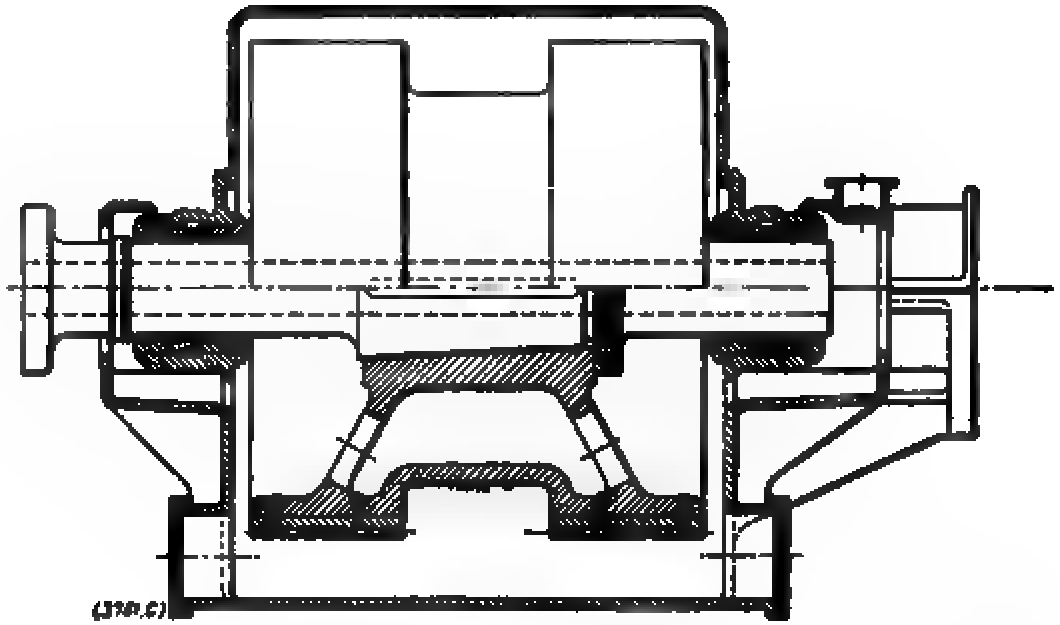
No. 103.—Cross Section of Teeth of Driving Wheel for Geared Marine Steam Turbine (Half Size).

Section taken Perpendicularly to Shaft.

on a suitable machine to produce automatically the correct tooth form of the pinions and the large wheel.

The turbines are designed to give their sea power at 1700 revs. per min., corresponding to 65 revs. per min. of the propeller. For astern driving, a turbine is embodied in the after end of the port side turbine, the latter being the low pressure turbine. As usual in such arrangements, the low pressure and astern turbines exhaust into the same connection, at about the middle of the length of the port turbine casing, so that when one of these is under steam the other runs idly in the vacuum. For compactness, the astern turbine has one impulse stage at its high pressure end.

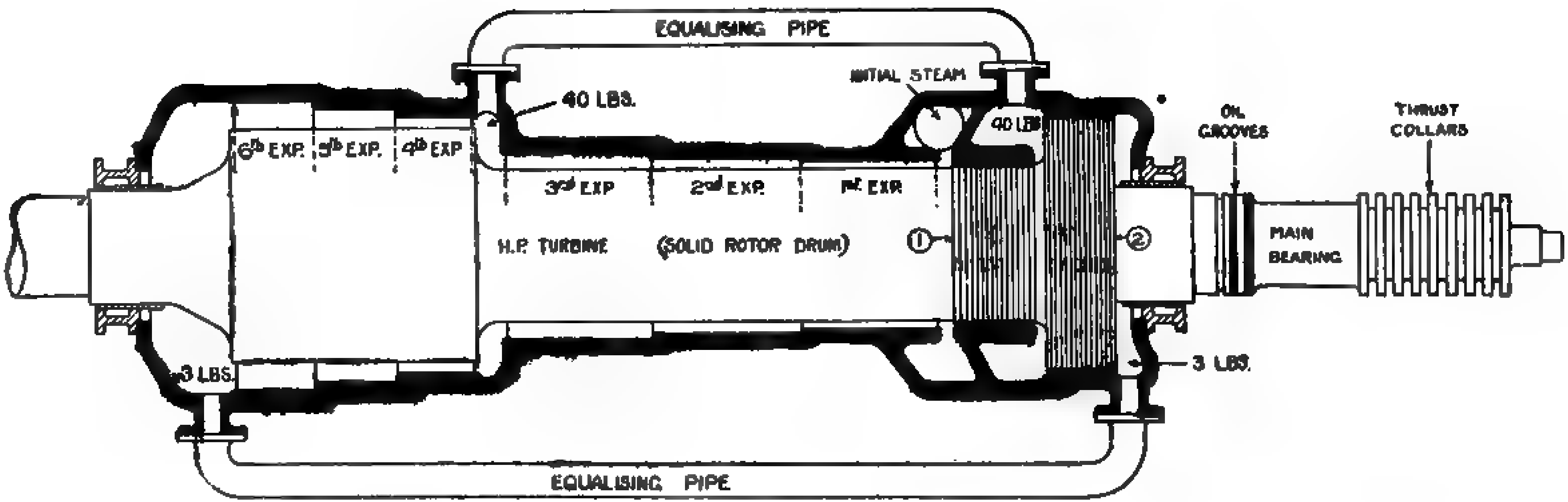
The lubrication of the main machinery is effected by oil fed down by gravity from a tank at about the level of the main deck. A thick carbon-filtered pure mineral oil is used. The same oil is used for the turbine bearings, gear bearings, and gear teeth, the oil



No. 108.—Gear Wheel, Pinions, and Gear Case, Channel Steamer "Paris."

Constructed by the Parsons' Marine Steam Turbine Co. Ltd.

Data.—Two driving shafts, each with one propeller.
„ H.P. turbines, each with six expansions.
„ L.P. „ „ „
„ astern „ „ five „ (in L.P. casings).
H.P. turbines geared down as 6 to 1.
L.P. „ „ „ 4·25 to 1.
Revolutions of driving shafts (2) 435 per minute.
„ H.P. turbines (6 × 435) = 2610 per minute.
„ L.P. „ (4·25 × 435) = 1848 „
Speed . . . . 25 knots.
S.H.P. . . . . 14000.



No. **Pistons and Equalising Pipes.**

Turbine of Geared-Down Set.

1. **After Dummy.** 2. **Forward Dummy.**

The rotor is of solid construction, and is fitted with two balance dummies placed at the forward end. The steam thrust aft is counterbalanced by means of the two pressure equalising pipes, one of which balances the pressure on the first three expansions of the smaller diameter of the rotor, and the other the pressure on the last three expansions of the larger diameter of rotor.

being delivered to the latter near the pinions by a series of jets. Oil from all these parts passes by gravity to a closed tank which is dropped into the double bottom, from which it is lifted by a steam-driven pump, passed through a filter, and through a cooler supplied with water from the main circulating pump up to the gravity tank again. The same oil is thus used over again, about 600 gallons being in circulation.

Data.—Two driving shafts, each with one propeller.
„ H.P. turbines, each with six expansions.
„ L.P. „ „ „
„ astern „ „ five „ (in L.P. casings).
H.P. turbines geared down as 6 to 1.
L.P. „ „ „ 4·25 „ 1.
Revolutions of driving shafts (2) 435 per minute.
„ H.P. turbines (6 × 435) = 2610 per minute.
„ L.P. „ (4·25 × 435) = 1848 „
Speed - - - - - - 25 knots.
S.H.P. - - - - - - 14000.

Dummy Pistons and Equalising Pipes (109).

With geared turbines the H.P. rotor drum is usually solid and stepped in two diameters, each stepping requiring a corresponding balance piston or dummy, and suitable equalising pressure pipes connecting the two positions. Two balance pistons and dummy packings are therefore fitted at the forward end, provided with pipe connections arranged as follows:—

One from end of 3rd or 4th expansion to forward side of after dummy.

One from exhaust end to forward side of forward dummy.

The diameter of the balance dummies is about equal to the mean diameter across the blades of the corresponding step diameter of rotor.

As the L.P. drums are usually hollow, equalising pipes are not required, owing to the fact that both ends of the turbine being open and in through connection the same vacuum pressure maintains at either end, so that by suitable arrangement of rotor and dummy diameter the necessary balance may be obtained.

The equalising pressures look after themselves, no hand adjustment being necessary.

Equalising pipes are therefore required to make up for the want of propeller thrust when the rotor drum is of the solid pattern.

The pressure balance obtained by the pipes in this way counteracts the effect of the steam pressure on the blades acting aft and the surface exhaust pressure acting on the exhaust end of the rotor. The equalising of the pressure prevents the rotor from altering its position longitudinally.

SECTION X.

PRACTICAL OPERATION OF TURBINE MACHINERY.

Heating Up.—The matter of warming up the turbines previous to starting is of vital importance, and should receive special attention from the engineers in charge.

1. Start running of the circulators and air pumps, and admit heating steam at about 10 lbs. pressure right through the turbines to the condensers, after first getting rid of the heated air from the boilers, etc.

2. Uniform heating of all parts should be aimed at, and during this process (for which about six hours should be allowed) the rotor should be moved round slowly by the turning gear for quarter turn every thirty minutes.

3. During heating up the various clearances should be carefully read and noted down. The turbine drains should be opened to get rid of condensed water, which, if allowed to remain, might result in blade stripping when starting up.

4. The pipe lines ahead and astern should also be heated up gradually by opening up the steam admission valves of same.

5. The forced lubrication system should be tested by running the oil to flood up the bearings, and then trying the test cocks on the main bearings for oil flow through each.

6. *Note.*—In some steamers a rather ingenious tell-tale arrangement is fitted which indicates when the oil flow is checked in the turbine bearings. The action is as follows:—The oil flow from the turbine bearings passes through a small float tank in which a metal float moves vertically on a small spindle. When the oil flow falls below a certain level, the float in dropping makes contact with a metal stop below, and by means of a bell battery and connections starts

the ringing of a small alarm bell, thus indicating to the engineer on water that the oil service is at fault.

7. Steam should also be turned on to the steam glands until the required pressure shows on the gauges.

8. In testing the turbines give steam to ahead for a few revolutions, then shut off and give steam to astern for a few revolutions.

9. Previous to reporting "ready" take dummy clearance.

10. Whenever possible work up to full power by degrees only.

11. When running up to speed required set gland steam and drains, the latter at mininum opening, as any excess opening means leakage.

12. Previous to getting under way start up oil pumps and obtain required pressure of oil (this varies from 25 lbs. to 45 lbs.). Examine sight glass for oil flow through bearings. (The sight glasses mentioned, unfortunately, soon become dirty and unreliable.)

13. When running astern be careful to take dummy clearance and make record of same. The dummy clearance is affected by the following:—

(a) Unequal expansion of rotor and casing.
(b) Amount of rotor "float" allowed for oil film at thrust.
(c) Effect of thrust pressure forward and steam pressure aft, and which varies under different speed or power conditions.

14. When running at any required speed or revolutions, it is best to open the minimum number of nozzle valves which will give sufficient steam for the purpose at the maximum boiler pressure.

It is bad practice to open more valves than actually necessary, and then reduce the pressure by throttling to keep down the speed or power to the required figure.

15. In changing speed or reversing it is advisable to work up or work down to the power and speed required gradually, so that the risk of accident due to sudden expansion or to sudden contraction of the rotor or casing may be eliminated.

16. For quick emergency stopping of the turbines when running ahead, steam can be admitted to the reverse end.

17. For any given number of nozzle valves open, the main steam, vacuum, and gland pressures should be noted. This will afford a guide to the operation of the turbines.

18. To allow for quick emergency changes from ahead to astern the dummy clearances should not be kept too fine.

19. All impulse nozzles in use are best to be kept full open, and all not in use tight shut.

20. A record should be kept of the revolution speed obtained by various nozzle openings.

21. With Curtis turbines the gland steam seals are worked at a pressure of about 3 lbs. Take accurate records of the oil temperatures when entering and leaving the main bearings, the maximum of which should not exceed about 130° Fahr.

22. The main bearing at initial end of turbine requires most attention, as the highest temperature exists there.

23. The importance of the gland steam seal on the L.P. and astern turbines should not be overlooked.

24. The efficiency of turbine machinery depends principally on the following:—

(*a*) High pressure at initial end.
(*b*) Vacuum in condenser end, and in idle turbines.
(*c*) Fine blade tip clearance.
(*d*) Tightness of glands against steam leakage out, and against air leakage in.
(*e*) Closed exhaust system.

It should be noted that leaky L.P. glands allow the admission of air, and therefore produce a reduced vacuum in the condenser.

25. When standing by after stopping, stop oil pumps, slow down circulating pump, and open heating up steam to turbines, to prevent cooling down and contraction of the working parts; also open drains from turbines.

26. Running ahead under closed exhaust system the auxiliary exhausts are usually led to the third expansion of the turbine, but when running astern or manœuvring the auxiliary exhausts require to be led to the condenser.

27. Turbine oil only is suitable for forced lubrication, as under pressure ordinary engine oil tends to form an emulsion.

28. The steam strainers require to be examined and cleaned at regular intervals.

29. The oil cooling service pipes, strainers, and other connections require regular attention.

30. After lifting turbine top half casings for examination, cleaning, or repair, before replacing same make careful examination for any articles left behind, such as chisels, hammers, loose blades, or packers, also turn rotor round slowly and examine closely for loose binding wires on the blading. The neglect of this has in many cases led to damage due to blade stripping, etc., when under way again, as in this respect turbines are by no means "fool-proof."

31. A record should be kept of the blade tip clearances, the blade side clearances, and, most necessary of all, of the dummy clearance, taken "cold," "heating up," and when "running" ahead and astern.

Note.—Recent turbine experience has proved the following:—

1. Fine dummy clearance is not so necessary as was originally understood, equal economy being obtainable with $\frac{50''}{1000}$ clearance as with $\frac{15''}{1000}$ clearance.

2. In practice it is advisable to allow an oil film clearance at the thrust rings of not less than $\frac{15''}{1000}$ to act as a counterbalance for any change in the relative position of rotor and casing, due to difference in temperature, which occurs when changing speed or power.

3. When ample boiler steam is available better results as regards power and speed are obtained with a condenser vacuum of 27 in. or even less, than with a vacuum of 28 in. or 29 in. as usually carried

This, it will be noted, contradicts the generally accepted theory that turbine efficiency is largely dependent upon a high degree of vacuum in the condensers.

In consequence of this the condensers of new turbines under construction are being made of smaller capacity than those of former designs.

SECTION XI.

DATA FROM ACTUAL PRACTICE.

1.—Pressures at Turbine Expansions.

Turbine expansion pressures as shown on gauges (taken specially for the author):—

A. H.P. Receiver Steam - - - 150 lbs. gauge.
Port L.P. Receiver Steam - - - 22 „
Starboard L.P. Receiver Steam - - 20 „

	Port L.P.	Starboard L.P.
At 4th Expansion -	9·5 in. vacuum	8·5 in. vacuum
„ 5th „ -	16 „	14 „
„ 6th „ -	20 „	18 „
Condenser vacuum -	28½ in.	28½ in.

Revolutions per minute. { Port - - 336; Centre - 336; Starboard - 328 } Mean = 333·3.

Sea temperature = 52°
Discharge „ = 85°
Hotwell „ = 78°

B. H.P. Receiver Steam - - 168 to 170 lbs. gauge.
Port L.P. Receiver Steam - - 26·5 lbs.
Starboard L.P. Receiver Steam - 24·5 „

	Port L.P.	Starboard L.P.
At 4th Expansion -	6 in. vacuum	4 in. vacuum
„ 5th „ -	14 „	12 „
„ 6th „ -	18 „	18 „
Condenser vacuum -	28·5 in.	29 in.

Revolutions per minute. { Port - - 341; Centre - 345; Starboard - 337 } Mean = 341.

Sea temperature = 50°
Discharge „ = 84°
Hotwell „ = 90°

C. H.P. Receiver Steam - - - 162 lbs. gauge.
Port L.P. Receiver Steam - - 28 „
Starboard L.P. Receiver Steam - - 26 „

Note.—The L.P. turbine initial pressure is practically the same as the last H.P. expansion terminal pressure, less 1 lb. or so.

Initial Pressures in L.P. Expansions.

No.	Port.	Starboard.
1	28 lbs. gauge	26 lbs. gauge
2	12 „	9 „
3	3 „	$2\frac{3}{4}$ „
4	5 in. vacuum	4 in. vacuum
5	18 „	14 „ {Gland "leak-off" connected here.
6	20 „	19 „
7	22 „	22 „
8	26 „	23 „
Terminal 8	28 „	28 „

Notice that the condenser vacuum pressure is practically the terminal pressure of the last L.P. expansion.

B.—Blading Data.

H.P. Turbine.

No. of Expansion.	No. of Blade Rows.	Blade Heights.	Blade Tip Clearance.	Pitch of Blade Rows (Longitudinally).	Diameter of Rotor Drum.	Length of Rotor Drum.
1st	18	$1\frac{5}{16}$ in.	·04 in. or $\frac{40}{1000}$ in.	$1\frac{1}{4}$ in.	6 ft. 4 in.	7 ft. 9 in.
2nd	17	$2\frac{1}{4}$,,		$1\frac{1}{4}$,,		
3rd	17	$2\frac{7}{8}$,,		$1\frac{5}{16}$,,		
4th	17	$3\frac{5}{8}$,,		$1\frac{3}{8}$,,		

I.P. Turbine.

No. of Expansion.	No. of Blade Rows.	Blade Heights.	Blade Tip Clearance.	Pitch of Blade Rows (Longitudinally).	Diameter of Rotor Drum.	Length of Rotor Drum.
1st	17	$4\frac{1}{2}$ in.	·04 in.	$1\frac{19}{32}$ in.	7 ft. 4 in.	9 ft. 9 in.
2nd	17	$5\frac{1}{2}$,,	·04 ,,	$1\frac{5}{8}$,,		
3rd	17	7 ,,	·045 ,,	$1\frac{11}{16}$,,		
4th	16	9 ,,	·045 ,,	2 ,,		

L.P. Turbines (2).

No. of Expansion.	No. of Blade Rows.	Blade Heights.	Blade Tip Clearance.	Pitch of Blade Rows (Longitudinally).	Diameter of Rotor Drum.	Length of Rotor Drum.
1st	8	3 in.	·06 in.	$1\frac{1}{4}$ in.	10 ft. 5 in.	9 ft.
2nd	8	$4\frac{1}{8}$,,	·06 ,,	$1\frac{5}{16}$,,		
3rd	8	$5\frac{5}{8}$,,	·07 ,,	$1\frac{5}{8}$,,		
4th	8	$7\frac{3}{4}$,,	·08 ,,	2 ,,		
5th	8	$10\frac{1}{2}$,,	·10 ,,	$2\frac{1}{8}$,,		
6th	8	$10\frac{1}{2}$,,	·10 ,,	$2\frac{5}{16}$,,		
7th	8	$10\frac{1}{2}$,,	·10 ,,	$2\frac{1}{2}$,,		

Astern Turbines (2).

(Impulse Wheel, also Reaction Blading.)

Impulse Wheel.	Blade Heights.	Impulse Wheel.	Blade Heights.
1st Blade row (moving) -	1 in.	5th Blade row (moving) -	$2\frac{5}{8}$ in.
2nd ,, ,, (fixed) -	$1\frac{7}{32}$,,	6th ,, ,, (fixed) -	$2\frac{29}{32}$,,
3rd ,, ,, (moving) -	$1\frac{25}{32}$,,	7th ,, ,, (moving) -	$3\frac{3}{32}$,,
4th ,, ,, (fixed) -	$2\frac{11}{32}$,,		

Astern Turbines (2).

Reaction Blading.

No. of Expansion.	No. of Blade Rows.	Blade Heights.	Blade Tip Clearance.	Pitch of Blade Rows (Longitudinally).	Diameter of Rotor Drum.	Length of Rotor Drum.
1st	10	1 in.	·055 in.	1 7/16 in.	7 ft.	5 ft. 5½ in.
2nd	10	2 "	·055 "	1 7/16 "		
3rd	10	4 "	·060 "	1½ "		
4th	10	4 "	·06 "	2 "		

C.—Turbine Data.

H.P. Turbine.

Rotor Drum 15¾ in. diameter forward, and 25¼ in. diameter aft.

	Six Expansions.	Blade Heights.	Number of Blade Rows.	Blade Section Number.	Type and Width of Blade.
Drum diameter, 15¾ in.	1st	11/16 in.	11	130 B	"Normal," ⅜ in. wide.
	2nd	1 "	11	"	"
	3rd	1⅜ "	11	"	"
Drum diameter, 25¼ in.	4th	13/16 in.	4	130 B	"Normal," ⅜ in. wide.
	5th	1⅛ "	4	"	"
	6th	1⅝ "	4	"	"

Notice that the H.P. drum is stepped up in two diameters.

L.P. Turbine.

Rotor Drum 40 in. diameter forward, and 37 in. diameter aft.

	Six Expansions.	Blade Heights.	Number of Blade Rows.	Blade Section Number.	Type and Size of Blades.
Drum diameter, 40 in.	1st	1¼ in.	3	130 B	"Normal" ⅜ in. wide
	2nd	1⅞ "	3	"	"
	3rd	2 3/16 "	3	"	"
Drum diameter, 37 in.	4th	4¼ in.	3	240 B	"Normal" ½ in. wide
	5th	7¼ "	3	"	"
	6th	7¼ "	3	"	"

Notice that the L.P. drum is stepped down in two diameters.

Astern Turbine (Impulse Wheel at 1st Stage).

Rotor Drum 26¼ in. diameter.

Four Expansions.	Blade Heights.	Number of Blade Rows.	Blade Section Number.	Type and Size of Blades.
1st	¾ in.	4	130 B	"Normal," ⅜ in. wide.
2nd	1½ ,,	4	,,	,,
3rd	3 ,,	4	240 B	"Normal," ½ in. wide.
4th	3 ,,	4	242 B	"Wing," ½ in. wide.

Note.—The blade tip clearance is neglected in the foregoing data: if allowed for, it will reduce the blades heights from about $\frac{20}{1000}$ or ·02 in. at the initial end to, say, $\frac{60}{1000}$ or ·06 at the exhaust end.

D.—Blading Data.

H.P. Turbine.

No. of Expansion.	No. of Blade Rows.	Blade Heights.	Pitch of Blade Rows (Longitudinally).	Diameter of Rotor Drum.	Length of Rotor Drum.
1	18	1⅜ in.	1¼ in.	5 ft. 8 in.	7 ft. 9 in.
2	16	2⅛ ,,	$1\frac{5}{16}$,,		
3	16	2¾ ,,	1⅜ ,,		
4	16	3½ ,,	$1\frac{7}{16}$,,		

I.P. Turbine.

1	15	4⅜ in.	1⅝ in.	5 ft. 10 in.	9 ft. 2 in.
2	15	5½ ,,	$1\frac{11}{16}$,,		
3	15	7 ,,	1¾ ,,		
4	15	9 ,,	2⅛ ,,		

L.P. Turbines (2).

1	8	3 in.	1¼ in.	8 ft. 4 in.	9 ft.
2	8	4⅛ ,,	$1\frac{5}{16}$,,		
3	8	5⅝ ,,	1⅝ ,,		
4	8	7¾ ,,	2 ,,		
5	8	10½ ,,	2⅛ ,,		
6	8	10½ ,,	$2\frac{5}{16}$,,		
7	8	10½ ,,	2½ ,,		

Astern Turbines (2).

1	9	$\frac{5}{8}$ in.	$1\frac{1}{16}$ in.	6 ft.	7 ft. 8 in.
2	9	$1\frac{1}{4}$ „	$1\frac{7}{16}$ „		
3	9	$2\frac{1}{4}$ „	$1\frac{7}{16}$ „		
4	9	$4\frac{1}{4}$ „	$1\frac{3}{4}$ „		
5	9	$4\frac{1}{4}$ „	$1\frac{7}{8}$ „		
6	9	$4\frac{1}{4}$ „	2 „		

	H.P.	I.P.	L.P.P.	L.P.S.
Dummy clearance cold - - -	·039 in.	·039 in.	·040 in.	·044 in.
„ „ heated up -	·045 „	·043 „	·059 „	·057 „
„ „ running ahead -	·046 „	·042 „	·059 „	·059 „

E.—Type—High Speed Craft.

S.H.P. - - - 21000.
Revolutions - - 350.
Speed - - - 25 knots.
Boiler steam - - 230 lbs.

Turbine arrangement -
- Two ahead turbines.
- „ astern „
- „ shafts, each with one propeller.
- Both ahead and astern turbines of the combined impulse and reaction type.

Impulse Blading.

(Ahead Turbines.)

Number of Rows.	Blade Heights.	Blade Height Increase.
1st Row, Moving	$1\frac{1}{4}$ in.	0
2nd „ Fixed	$1\frac{17}{32}$ „	$\frac{9}{32}$ in.
3rd „ Moving	$1\frac{13}{16}$ „	$\frac{9}{32}$ „
4th „ Fixed	$2\frac{3}{32}$ „	$\frac{9}{32}$ „
5th „ Moving	$2\frac{3}{8}$ „	$\frac{9}{32}$ „
6th „ Fixed	$2\frac{21}{32}$ „	$\frac{9}{32}$ „
7th „ Moving	$2\frac{27}{32}$ „	$\frac{6}{32}$ „

Blade tip clearance, $\frac{3}{32}$ in. or $\frac{93}{1000}$ in.
Impulse wheel diameter about 9 ft.

Reaction Blading.
(Ahead Turbines.)

Number of Expansion.	Number of Blade Rows.	Blade Heights.	Blade Pitch (Longitudinally).	Blade Tip Clearance.
1	9	$1\frac{1}{8}$ in.	$1\frac{1}{4}$ in.	·06
2	9	$1\frac{5}{8}$ „	$1\frac{5}{16}$ „	·065
3	9	$2\frac{3}{8}$ „	$1\frac{3}{8}$ „	·07
4	9	$3\frac{1}{2}$ „	$1\frac{7}{16}$ „	·075
5	9	$5\frac{1}{4}$ „	$1\frac{13}{16}$ „	·08
6	9	$7\frac{1}{2}$ „	$1\frac{7}{8}$ „	·09
7	9	$7\frac{1}{2}$ „	$2\frac{1}{16}$ „	·09
8	4	$7\frac{1}{2}$ „	$2\frac{3}{16}$ „	·09
9	4	$7\frac{1}{2}$ „	$2\frac{5}{16}$ „	·09

Rotor diameter, 8 ft. 3 in.

Impulse Blading.
(Astern Turbines.)

Number of Rows.	Blade Heights.	Blade Height Increase.
1st Row, Moving	$1\frac{11}{32}$ in.	0
2nd „ Fixed	$1\frac{25}{32}$ „	$\frac{7}{16}$ in.
3rd „ Moving	$2\frac{3}{32}$ „	$\frac{5}{16}$ „
4th „ Fixed	$2\frac{13}{32}$ „	$\frac{5}{16}$ „
5th „ Moving	$2\frac{23}{32}$ „	$\frac{5}{16}$ „
6th „ Fixed	$3\frac{1}{32}$ „	$\frac{5}{16}$ „
7th „ Moving	$3\frac{7}{32}$ „	$\frac{3}{16}$ „

Blade tip clearance, $\frac{3}{32}$ in. or $\frac{93}{1000}$ in. (·093″).
Impulse wheel diameter about 8 ft.

Reaction Blading.
(Astern Turbines.)

Number of Expansion.	Number of Blade Rows.	Blade Heights.	Blade Pitch (Longitudinally).	Blade Tip Clearance.
1	4	$1\frac{1}{8}$ in.	$1\frac{3}{8}$ in.	·06
2	4	$2\frac{1}{4}$ „	$1\frac{3}{8}$ „	·07
3	4	$4\frac{1}{2}$ „	$1\frac{3}{4}$ „	·08
4	4	$4\frac{1}{2}$ „	2 „	·08

Rotor diameter about 7 ft.

F.—Blading Data.

H.P. Turbines.

	Six Expansions.	Number of Blade Rows.	Blade Heights.
Rotor drum, 14 in. diameter.	1st	11	$1\frac{5}{16}$ in.
	2nd	11	$1\frac{7}{8}$,,
	3rd	11	$2\frac{5}{8}$,,
Rotor drum, 23 in. diameter.	4th	4	$1\frac{5}{8}$,,
	5th	4	$2\frac{1}{4}$,,
	6th	4	$3\frac{1}{4}$,,

Rotor stepped in two diameters, 14 in. and 23 in.

Note.—The actual blade heights are less than above by the tip clearance, which reduces the effective heights by an amount ranging from ·030 in. to ·06 in. according to position.

L.P. Turbines.

Six Expansions.	Number of Blade Rows.	Blade Heights.
1st	3	$2\frac{3}{8}$ in.
2nd	3	$3\frac{1}{8}$,,
3rd	3	$4\frac{3}{8}$,,
4th	3	6 ,,
5th	3	6 ,, semi-wing
6th	3	6 ,, wing

Rotor drum, 37 in. diameter.

Astern Turbines.

(Reaction Blading.)

Four Expansions.	Number of Blade Rows.	Blade Heights.
1st	4	$1\frac{1}{16}$ in.
2nd	4	$2\frac{1}{8}$,,
3rd	4	$4\frac{1}{4}$,,
4th	4	$4\frac{1}{4}$,,

Rotor drum, 24 in. diameter.

The first stage of the astern turbine consists of an impulse wheel, 40 in. diameter, and containing three moving blade rows and two fixed blade rows, the blade heights ranging from 1 in. to $2\frac{1}{4}$ in. approximately.

The strong blading of the impulse stage permits of the full boiler steam being suddenly applied for quick reversing without much danger of blade-stripping taking place.

G.—Type—High Speed Craft.

S.H.P. - - 12000. Revs. - - 600.

Turbine Arrangement.—Combined impulse and reaction turbines. Two ahead turbines and two astern turbines on each shaft, each consisting of impulse nozzles and bucket wheel, also reaction blading in series with same. Two shafts with one propeller to each. The turbines expand the initial steam of 220 lbs. pressure down to the condenser pressure direct, and are therefore both high and low pressure combined.

Impulse Blading.

Ahead Turbine.

		Blade Heights.	Increase in Blade Heights.
Ahead turbine 4 Moving Rows, 3 Fixed Rows.	1st Blade Row, Moving -	$1\frac{11}{16}$ in.	0
	2nd " " Fixed -	$2\frac{3}{16}$ "	$\frac{1}{2}$ in.
	3rd " " Moving -	$2\frac{5}{8}$ "	$\frac{7}{16}$ "
	4th " " Fixed -	$3\frac{1}{8}$ "	$\frac{1}{2}$ "
	5th " " Moving -	$3\frac{5}{8}$ "	$\frac{1}{2}$ "
	6th " " Fixed -	$4\frac{1}{8}$ "	$\frac{1}{2}$ "
	7th " " Moving -	$4\frac{5}{16}$ "	$\frac{3}{16}$ "

Reaction Blading.

Ahead Turbine.

No. of Expansion.	Blade Rows.	Blade Heights.	Pitch of Blades (Longitudinally).
1	9	$1\frac{9}{16}$ in.	$1\frac{1}{4}$ in.
2	9	$2\frac{1}{2}$ "	$1\frac{1}{4}$ "
3	8	4 "	$1\frac{5}{16}$ "
4	8	$6\frac{1}{8}$ "	$1\frac{11}{16}$ "
5	8	10 "	2 "
6	8	10 "	$2\frac{3}{16}$ "
7	6	10 "	$2\frac{3}{8}$ "
8	4	10 "	$2\frac{3}{4}$ "

Mean diameter of impulse wheel = 6 ft. 4 in.
Diameter of rotor drum - - = 5 ft.

Notice that the last four expansions are of equal blade height, but are of increasing longitudinal pitch. This is to allow for the increase of blade angles which cant the blades round more in a fore and aft line, and thus take up more room.

H.—Type—High Speed Passenger Steamer.

General Data.

3 ahead turbines.
2 astern „ (in L.P. casings).
3 shafts.

S.H.P. - - -	16800 (total).
Speed - - -	20·5 knots.
Revs. per min. - -	port, 372; centre, 368; starboard, 356 } mean = 366.
Boiler steam - -	180 lbs. gauge.
H.P. initial steam - -	170 „
H.P. terminal steam -	27 „
Port L.P. initial steam -	25 „
„ terminal pressure -	27 in. vacuum.
Start L.P. initial steam -	25 lbs. gauge.
„ terminal pressure	27 in. vacuum.
Condenser vacuum - -	$27\frac{1}{2}$ in. to 28 in.
Astern turbines - -	24 in. vacuum.
H.P. gland steam - -	1 to 2 lbs. forward and aft.
L.P. gland steam - -	2 to 4 „ „
Oil pressure - - -	15 lbs.
Propeller diameter - -	8 ft. $1\frac{1}{2}$ in.
„ pitch - -	8 ft. $1\frac{1}{2}$ in.
„ expanded area -	28·5 sq. ft. (three blades).
„ projected area -	26·5 „ „
Coal per H.P. hour - -	1·26 lbs.

I.—Type—Fast Passenger Steamer.

6 turbines in all.

Turbine arrangement {
1 H.P. turbine.
1 I.P. „
2 L.P. „
2 astern turbines (in L.P. casings).

The astern turbines are of the Parsons' combined impulse and reaction type, and are not fitted with nozzle control valves; all other turbines are of reaction type only.

S.H.P. - - - -	20700.
Speed - - - -	20 knots.
Slip - - - - -	18·6 per cent.
Propeller pitch - - -	8′ 6″.
Mean revs. - - - -	305.
Boiler steam - - - -	193 lbs. gauge.
Engine steam - - -	190 „
H.P. steam - - - -	184 „
I.P. „ - - - -	41 „
L.P.P. „ - - - -	4·5 „
L.P.S. „ - - - -	3·5 „
Vacuum - - - -	27·4 in.
Barometer - - - -	29·4 „
Coal per S.H.P. per hour all purposes	1·23 lbs.
Temperature of oil at delivery from bearings - - - -	175° F.

Note.—With both L.P. turbines running only, the speed obtained was 10 knots, the L.P. initial pressure being 15″ vacuum and the terminal pressure 29″ vacuum.

Steam Connections.

Each of the four ahead turbines have boiler steam connections for independent working. The H.P. exhaust is led to the I.P. for ordinary running, but separate exhaust to each L.P. turbine is also arranged for, in case of breakdown. The various combinations of turbine connections may be summarised as follows:—

1. H.P. exhaust to I.P., and I.P. exhaust to each L.P., thence to condensers.

2. H.P. exhaust to both L.P. turbines, or to one only, thence to condenser. I.P. cut out.

3. I.P. exhaust to both L.P. turbines, or to one only, thence to condensers. H.P. cut out.

4. L.P. exhaust to condensers direct, H.P. and I.P. cut out.

5. Reverse steam exhaust to condensers; all ahead turbines running idle.

Note.—In ordinary ahead running the auxiliary exhausts are lead to the L.P. turbines.

J.—Type—Passenger Cargo Steamer.

Speed 17·5 Knots. S.H.P. 11000

Two shafts, each connected up by gear wheels to one H.P. and one L.P. turbine.

Four ahead reaction turbines.

Two astern impulse and reaction turbines; each astern turbine contained in L.P. casing, but placed forward instead of aft.

Gear down ratio as 12·5 : 1.

One marine type thrust block to each main shaft.

Flexible couplings fitted between turbine spindles and pinion wheel shaft within gear case.

H.P. turbines each placed centre.

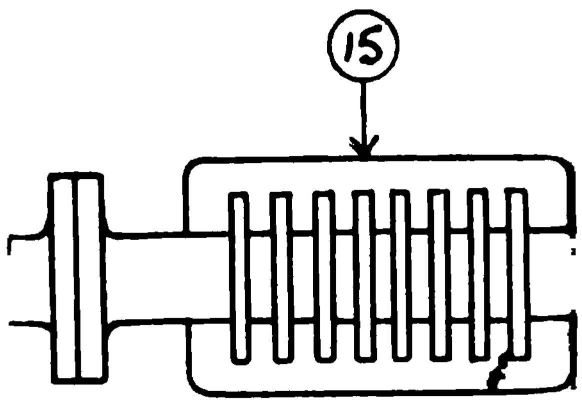
L.P. " " port and starboard.

Turbine Pressures, etc.

	Boiler Steam (Gauge).	H.P. Turbine Steam.		H.P. Turbine 3rd Expansion Steam.		L.P. Turbine Steam.		L.P. Turbine Exhaust.	
		Port.	Starboard.	Port.	Starboard.	Port.	Starboard	Port.	Starboard.
	Lbs.							In.	In.
No. 1	196	178	173	110	104	1·5	5	27·9	27·9
No. 2	188	171	166	105	101	·70	3	28·3	27·8

Vacuum in Condensers.		Barometer.	Revolutions.		Shaft Horse Power.		Speed in Knots.	Weather Conditions.
Port.	Starboard.		Port.	Starboard.	Port.	Starboard.		
In.	In.	In.						
28	28	29·4	114	118	4100	4465	13·5	Heavy sea and wind.
28·35	28	30	120	121	3865	4170	16·25	Moderate sea

Temperature of Bearings.	L.P. Forward.	L.P. Aft.	Gear Case H.P.F.	Gear Case L.P.F.	L.P. Aft.	Tunnel Bearings.
Inlet - - -	112°	102°	105°	102°	102°	90°
Outlet - -	122°	108°	108°	108°	108°	98°



1. H.P. tu
2. H.P. ex
3. H.P. ex
4. L.P. ah
5. L.P. as
6. Control
7. Auxiliar
8. Strainer

"The Marine Steam Turbine."

same time. Boiler steam direct to L.P
arranged for as shown. A throttle val
turbine governors is fitted in the inlet b

No. 114.—Manœuvring Valves.

"Transylvania" and "Tuscania."

One hand wheel controls boiler steam to the ahead and astern turbines. The slotted lever shown acts to lock the gear and so prevent steam from passing to ahead and astern turbines at the same time. Boiler steam direct to L.P. ahead turbines is also arranged for as shown. A throttle valve connected up to the turbine governors is fitted in the inlet branch of the chest.

K.—Type—Fast Passenger Steamer.

Geared-Down Turbines.

(Two shafts, two H.P. turbines, two L.P. Turbines, and two astern turbines.)

Horse power	12000.
Speed	22·5 knots.
Propeller pitch	10 ft.
„ diameter	8 ft. 6 in.
Revolutions P	295.
„ S	297.
Boiler steam	197 lbs. gauge.
H.P. turbine steam P	185 „
„ „ S	180 „
L.P. steam P	1 „
„ S	·8 „
Vacuum P	29 in.
„ S	29 „
Gland pressure H.P.	2 lbs.
„ „ L.P.	0 „
Feed pump pressure	238 „
Filter pressure	238 „
Oil pressure in bearings, etc.	50 „
Slip per cent.	20 per cent.
Diameter H.P. rotor spindles	5½ in.
„ L.P. „ „	6¾ „
„ tunnel shaft	10½ „
„ propeller „	11⅝ „
Centres of tunnel shafts	15 ft. aft, and 12 ft. 6 in. forward.

Pressures in H.P. Expansions.

The following record of pressures existing in the eight H.P. turbine expansions are of special interest.

No. of Expansion (H.P.).	Initial Pressure.
1st Expansion	173 lbs. gauge.
2nd „	132 „
3rd „	99 „
4th „	69 „
5th „	48 „
6th „	31 „
7th „	24 „
8th „	14 „
8th „ Terminal Pressure	6 „

Turbine Gearing.

The turbine set consists of six turbines in all, four ahead and two astern, connected to two shafts, each having a single propeller. Each shaft is driven by one H.P. and one L.P. turbine geared down in speed in the following ratio:—

H.P. gear as 9·3 : 1.
L.P. „ 7 : 1.

So that at, say, 285 revolutions of propelling shafts, the turbine revolutions would be as follows:—

H.P. = 9·3 × 285 = 2650 revolutions per minute.
L.P. = 7 × 285 = 1995 „ „

Details of Gear Wheels.

	Diameter of Pitch Circle.	Number of Teeth.
Gear wheel - -	65·7 ins.	253
H.P. pinion - -	7·0 „	27
L.P. „ - -	9·3 „	36

H.P. Ratio = 253 ÷ 27 = 9·3.
L.P. „ = 253 ÷ 36 = 7.

Note.—Circumferential pitch of teeth = ·8159 in.

Centres of Turbines and Gear Wheel.

H.P. centre to gear wheel centre = 36⅝ in.
L.P. „ „ „ = 37½ „

L.—Type—Fast Passenger Steamer.

Six turbines in all, consisting of:—

Turbine arrangement: 1 H.P.; 1 I.P.; 2 L.P.'s.; 2 astern (in L.P. casings.)

S.H.P. (total) - - - - 27280.
Speed - - - - 21·43 knots.
Slip - - - - - 20·6 per cent.

Revs. H.P.	326	Mean revs. = 332.
Revs. I.P.	344	
Revs. L.P.P.	333	
Revs. L.P.S.	327	
Boiler steam	190 lbs. gauge.	
H.P. „	177 „	
I.P. „	62 „	
L.P.P. „	11·5 „	
L.P.S. „	11·4 „	
Vacuum	27·6 ins.	
Barometer	29·46 „	
Draught at air trunks	1·6 „	
Draught below bars	$\frac{3}{4}$ „	
Draught above „	$\frac{3}{16}$ „	
No. of boilers	10.	
„ furnaces	64.	
Coal per square foot of grate per hour	20·57 lbs.	
Coal per S.H.P. per hour (propelling purposes)	1·24 lbs.	

M.—U.S. Destroyers "Sterrett" and "Perkins."

(Curtis Turbines.)

Twin screw—Two ahead turbines, and two astern turbines.

Speed	29·7 knots.
Revolutions	593·5.
Nozzles open	12.
Initial pressure	231 lbs. gauge.
Dryness	·978.
First stage pressure	71 lbs. gauge.
Vacuum	27·3 in.
Barometer	30 in.
S.H.P. (each ahead turbine)	5834.
Pounds water per H.P. hour (all purposes)	14·49.
Blade speed	187 ft. per second.
Calculated nozzle discharge steam speed	2080 ft. per second.
Angle of nozzle centre line	20°.
Number of nozzles at 1st stage	19.
Mean diameter across blades	6 ft.
Number of ahead stages	14 (each one "velocity compounded").
Total throat area of nozzle (1st stage)	8·89 sq. in.
Total area at end of taper	10·17 sq. in.

All wheel stages have partial circumferential admission, the lower pressure ones having admission at both top and bottom of casing. The drum stages have all complete circumferential admission.

The drum section consists of eight stages, each one velocity compounded by means of two moving blade rows.

1st stage - - -	4 Moving blade rows. 3 Fixed " "
Stages, 2, 3, 4, 5, 6 - -	3 Moving " " 2 Fixed " "
Stages, 7, 8, 9, 10, 11, 12, 13, 14	2 Moving " " 1 Fixed " "

1st Stage Depth of Nozzle Opening, 1·2 in.	Effective Blade Heights.	Blade Angles.		No. of Blades per Row.
		Admission Edge.	Exit Edge.	
1st Row, Moving - -	1·33 in.	28°	22°	468
2nd " Fixed - - -	1·65 "	28°	22°	468
3rd " Moving - -	1·98 "	34°	22°	468
4th " Fixed - - -	2·23 "	34°	24°	566
5th " Moving - -	2·49 "	42°	23°	566
6th " Fixed - - -	2·75 "	42°	23°	566
7th " Moving - -	3 "	68°	45°	566

2nd Stage to 6th Stage. Depth of Nozzle Openings, 1·25 in.	Effective Blade Heights.	Blade Angles.		No. of Blades per Row.
		Admission Edge.	Exit Edge.	
1st Row, Moving - -	1·54 in.	28°	23°	566
2nd " Fixed - - -	1·83 "	34°	24°	566
3rd " Moving - -	2·12 "	40°	30°	566
4th " Fixed - - -	2·41 "	53°	29°	566
5th " Moving - -	2·70 "	78°	45°	566

Astern Turbine.

Consists of two wheel stages, each stage carrying 4 moving blade rows and 3 fixed blade rows.

Area of Nozzle at throat = ·675 sq. in.
" " end of taper = 1·185 "

N.—Entropy Diagram from Actual Data.

The entropy diagram (115) which follows has been developed from actual data from practice, and the difference in B.T.U. drop observable in each turbine expansion as worked out would probably be caused by inequalities in the blading or other constructional causes.

DATA.

S.H.P. = 16800.
Water consumption = 14 lbs. per S.H.P. per hour.
Ahead { 1 H.P. turbine with 4 expansions.
{ 2 L.P. " 8 "

INITIAL PRESSURES IN EXPANSIONS AS SHOWN BY GAUGES.

	Expansion.	Gauge Pressure.	Absolute Pressure.	Temp. Fahr. (From Table).
				Degrees.
H.P. turbine	1	165 lbs.	180 lbs.	373
	2	105 "	120 "	341
	3	65 "	80 "	312
	4	40 "	55 "	287
L.P. turbine	1	25 "	40 "	267
	2	11 "	26 "	242
	3	3 "	18 "	222
	4	6 in. vac.	12 "	202
	5	15 "	7·5 "	180
	6	20 "	5 "	162
	7	23 "	3·5 "	147
	8	25.4 "	2.3 "	132
Terminal pressure	8	27 "	1.5 "	116

For approximate results only the atmospheric pressure has been taken as equal to 15 lbs. per square inch, or, say, 30 in. barometer, so that to find the absolute pressure at, for example, the 7th L.P. expansion—

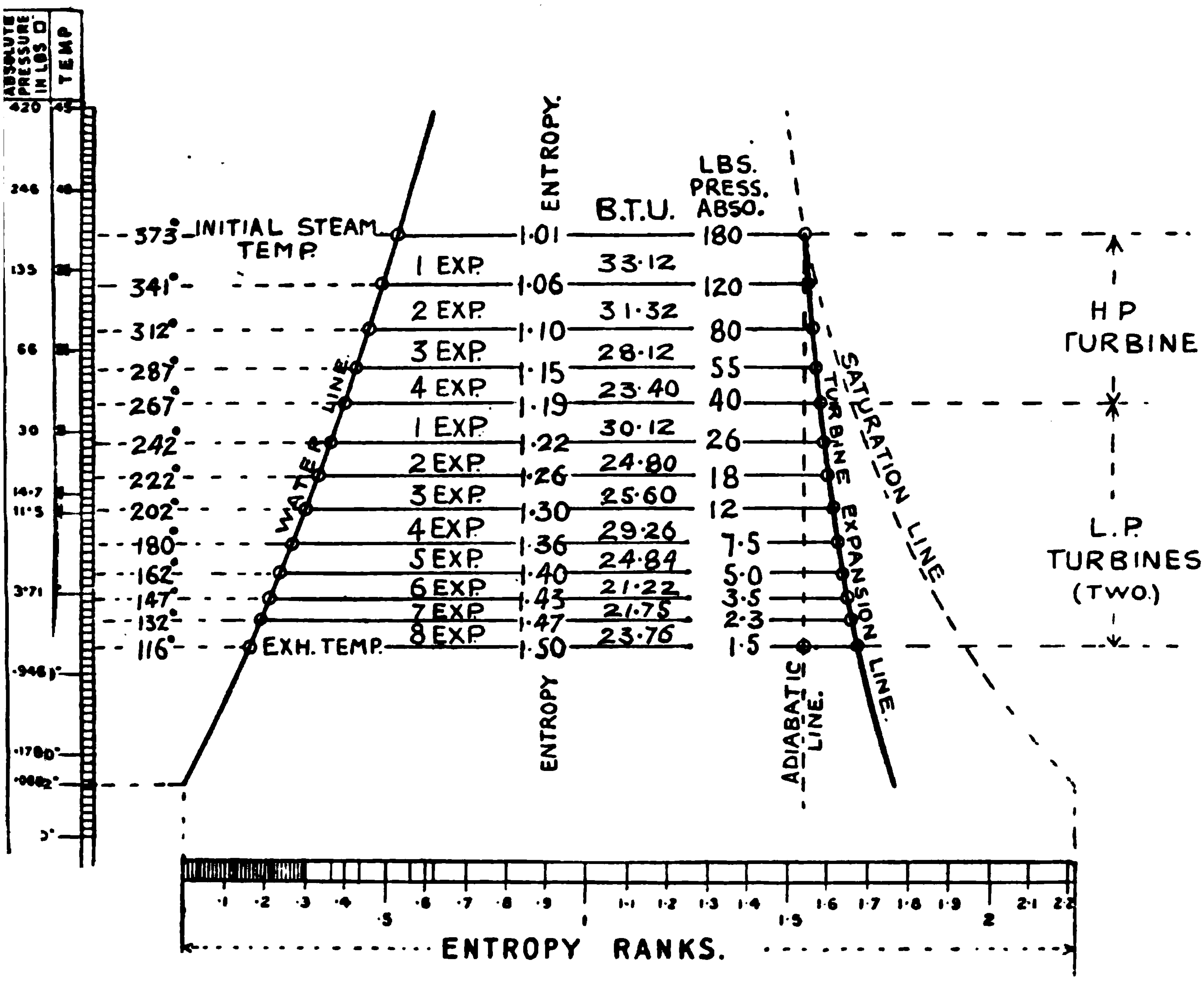
Then, $$\text{Absolute pressure} = \frac{30 \text{ in.} - 23 \text{ in.}}{2} = 3{\cdot}5 \text{ lbs.}$$

Or, at the 8th expansion,

Then, $$\text{Absolute pressure} = \frac{30 \text{ in.} - 25{\cdot}4 \text{ in.}}{2} = 2{\cdot}3 \text{ lbs.}$$

Construction of Diagram.

On the usual entropy diagram (115) set off the turbine expansion line which, in the present case, has been assumed to occupy a position one-third of the distance between the adiabatic and saturation lines as shown.



No. 115.—Entropy Diagram.

(Data from Actual Practice.)

Next draw horizontal entropy lines at the various temperature levels corresponding to the pressures shown in the turbine expansions, H.P. and L.P., and measure across the entropy values between the water line on left, and the turbine expansion line on right, marking these down as found and shown on the diagram, then for each expansion the B.T.U. drop is equal to :—Mean entropy × Temperature difference = B.T.U.

H.P. Turbines.

1st Expansion B.T.U. $= \left(\frac{1 \cdot 01 + 1 \cdot 06}{2}\right) \times (373^\circ - 341^\circ) = 33 \cdot 12$ B.T.U.

2nd „ B.T.U. $= \left(\frac{1 \cdot 06 + 1 \cdot 10}{2}\right) \times (341^\circ - 312^\circ) = 31 \cdot 32$ „

3rd „ B.T.U. $= \left(\frac{1 \cdot 10 + 1 \cdot 15}{2}\right) \times (312^\circ - 287^\circ) = 28 \cdot 12$ „

4th „ B.T.U. $= \left(\frac{1 \cdot 15 + 1 \cdot 19}{2}\right) \times (287^\circ - 267^\circ) = 23 \cdot 40$ „

L.P. Turbines.

1st Expansion B.T.U. $= \left(\frac{1 \cdot 19 + 1 \cdot 22}{2}\right) \times (267^\circ - 242^\circ) = 30 \cdot 12$ B.T.U.

2nd „ B.T.U. $= \left(\frac{1 \cdot 22 + 1 \cdot 26}{2}\right) \times (242^\circ - 222^\circ) = 24 \cdot 80$ „

3rd „ B.T.U. $= \left(\frac{1 \cdot 26 + 1 \cdot 30}{2}\right) \times (222^\circ - 202^\circ) = 25 \cdot 60$ „

4th „ B.T.U. $= \left(\frac{1 \cdot 30 + 1 \cdot 36}{2}\right) \times (202^\circ - 180^\circ) = 29 \cdot 26$ „

5th „ B.T.U. $= \left(\frac{1 \cdot 36 + 1 \cdot 40}{2}\right) \times (180^\circ - 162^\circ) = 24 \cdot 84$ „

6th „ B.T.U. $= \left(\frac{1 \cdot 40 + 1 \cdot 43}{2}\right) \times (162^\circ - 147^\circ) = 21 \cdot 22$ „

7th „ B.T.U. $= \left(\frac{1 \cdot 43 + 1 \cdot 47}{2}\right) \times (147^\circ - 132^\circ) = 21 \cdot 75$ „

8th „ B.T.U. $= \left(\frac{1 \cdot 47 + 1 \cdot 50}{2}\right) \times (132^\circ - 116^\circ) = 23 \cdot 76$ „

Total B.T.U. available (per pound steam) $= 33 \cdot 12 + 31 \cdot 32 + 28 \cdot 12 + 23 \cdot 40 + 30 \cdot 12 + 24 \cdot 80 + 25 \cdot 60 + 29 \cdot 26 + 24 \cdot 84 + 21 \cdot 22 + 21 \cdot 75 + 23 \cdot 76 = 317 \cdot 31$.

Each S.H.P. uses 14 lbs. steam per hour.

Then, Efficiency = Work done ÷ Heat supplied.

Work done $= 33000 \times 60 = 1980000$ ft.-lbs.

Heat supplied $= 14 \times 317 \cdot 31 \times 778 = 3456138$ ft.-lbs.

So that, Turbine efficiency $= 1980000 \div 3456138 = \cdot 573$ or $57 \cdot 3$ per cent.

O.—Report on Working of Turbine Drains and Steam Glands.

"**Drains.**—The drain from the H.P. to L.P. is opened when the H.P. turbine is stopped and shut, except for one turn, when the H.P. turbine is steaming.

It is constantly open one turn to drain off any water which may accumulate. In port it is opened up direct to the bilges when the air pumps are stopped.

There is also a drain with two valves on the forward end of the H.P. to drain water from the forward end to the after end of the H.P. or direct to the bilges.

The one on after end is opened when starting to warm up, and is shut when all the water is out of the pipes and connections."

Steam Glands.—H.P. gland steam under ordinary working conditions acts as a leak-off to L.P. casing, but when working at low

pwers a partial vacuum forms in the gland pockets, owing to the rduced pressure then obtaining. In this case the "steam to glands" onnection is opened and the "leak-off" connection shut.

END ELEVATION–LOOKING AFT.

No. 116.

End View of turbines showing H.P. gland "leak off" and "steam to glands." The steam connection to L.P. glands is also shown, and the H.P. and L.P. cross drain connection.

Blade Side Clearance.

(Taken Cold.)

P.—Brown-Curtis Impulse Turbines.

The side clearance or longitudinal clearance between the blade rows fixed and moving is generally more on the forward side than on the after side, and ranges as follows :—

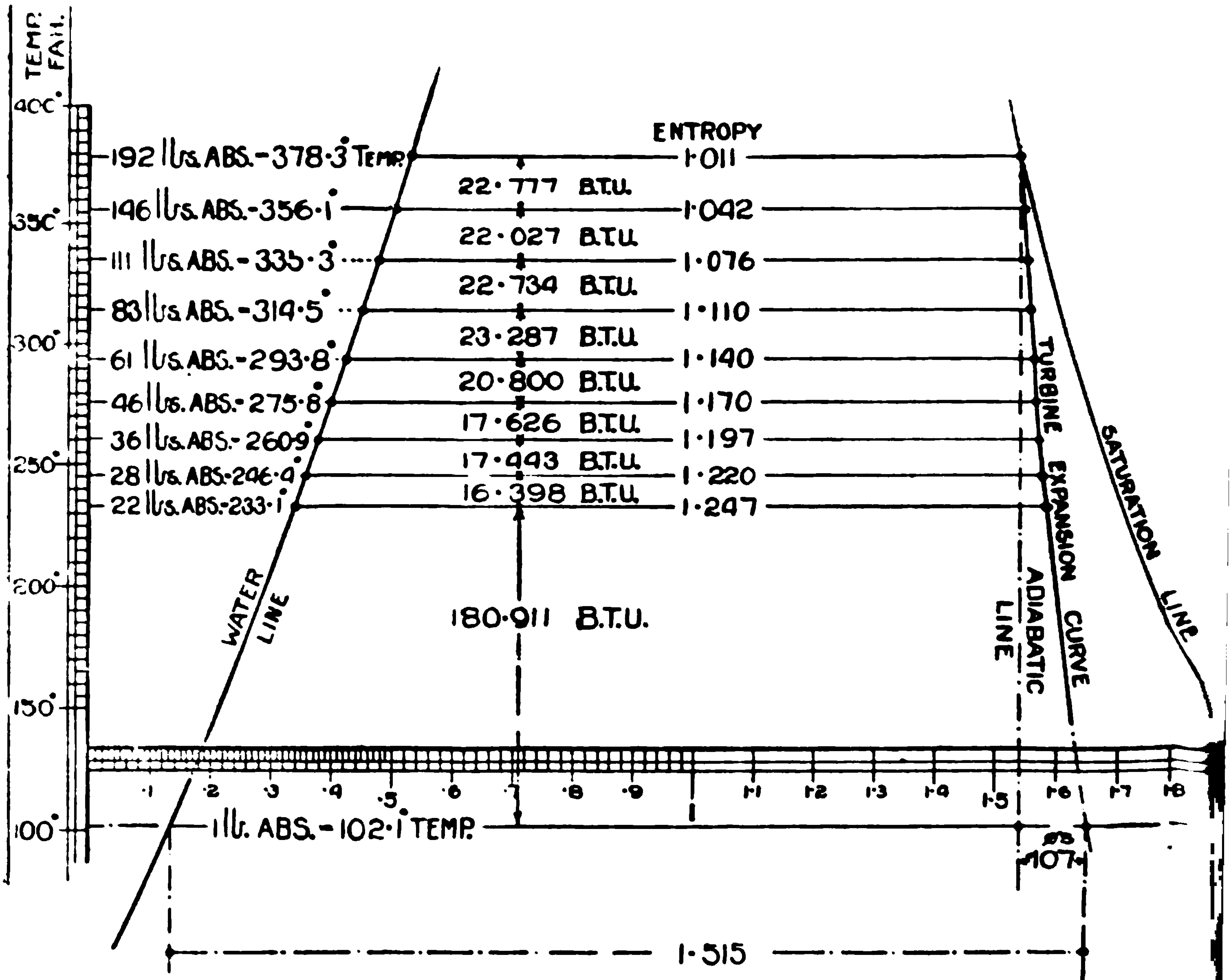
		Port.		Starboard.	
		Forward.	Aft.	Forward.	Aft.
1st Stage	1st Row -	...	183	...	·255
	2nd " -	·255	·232	·198	·276
	3rd " -	.265	·222	·202	·269
	4th " -	·246	·286	·218	·260
	5th " -	·250	·206	·230	·244
	6th " -	·240	·236	·260	·226
	7th " -	·301	...	·233	...

Notice that no clearance is given forward for the 1st row, nor aft for the last row of blades, as these rows are next to the steam nozzles of the 1st and 2nd stage respectively.

Q.—Entropy Diagram for H.P. Turbine with Eight Expansions.

(Data from Actual Practice.)

In this case the reheating effects of blade friction loss has been assumed to increase the heat area of the diagram by ·104 entropy,



No. 117.—Entropy Diagram.

(Data from Actual Practice.)

which, measured off to the right of the adiabatic line at the exhaust temperature level of 102°, gives the position of the "turbine expansion curve." As before described, draw horizontal lines at the various temperature levels corresponding to the absolute pressures shown on the turbine expansion gauges, and which, tabulated, read as follows:—

Expansion.	Initial Pressure. (Gauge).	Initial Pressure (Absolute).	Temperature Fahr.	Entropy.
			Degrees.	
1st - - -	177	192	378·3	1·011
2nd - -	131	146	356·1	1·042
3rd - - -	96	111	335·3	1·076
4th - - -	68	83	314·5	1·110
5th - - -	46	61	293·8	1·140
6th - - -	31	46	275·8	1·170
7th - - -	21	36	260·9	1·197
8th - - -	13	28	246·4	1·220
	(Terminal)			
8th - - -	7	22	233·1	1·247

Then, Heat Drop in B.T.U. = Mean Entropy × Difference of Temperature.

1st EXPANSION.

B.T.U.

$$\text{B.T.U.} = \left(\frac{1{\cdot}011 + 1{\cdot}042}{2}\right) \times (378{\cdot}3^\circ - 356{\cdot}1^\circ) = 22{\cdot}78.$$

2nd EXPANSION.

$$\text{B.T.U.} = \left(\frac{1{\cdot}042 + 1{\cdot}076}{2}\right) \times (356{\cdot}1^\circ - 335{\cdot}3^\circ) = 22{\cdot}02.$$

3rd EXPANSION.

$$\text{B.T.U.} = \left(\frac{1{\cdot}076 + 1{\cdot}110}{2}\right) \times (335{\cdot}3^\circ - 314{\cdot}3^\circ) = 22{\cdot}95.$$

4th EXPANSION.

$$\text{B.T.U.} = \left(\frac{1{\cdot}110 + 1{\cdot}140}{2}\right) \times (314{\cdot}3^\circ - 293{\cdot}8^\circ) = 23{\cdot}06.$$

5th EXPANSION.

$$\text{B.T.U.} = \left(\frac{1{\cdot}140 + 1{\cdot}170}{2}\right) \times (293{\cdot}8^\circ - 275{\cdot}8^\circ) = 20{\cdot}79.$$

6th EXPANSION.

$$\text{B.T.U.} = \left(\frac{1{\cdot}170 + 1{\cdot}197}{2}\right) \times (275{\cdot}8^\circ - 260{\cdot}9^\circ) = 17{\cdot}62.$$

7th EXPANSION.

$$\text{B.T.U.} = \left(\frac{1{\cdot}197 + 1{\cdot}220}{2}\right) \times (260{\cdot}9^\circ - 246{\cdot}4^\circ) = 17{\cdot}51.$$

8th EXPANSION.

$$\text{B.T.U.} = \left(\frac{1{\cdot}220 + 1{\cdot}247}{2}\right) \times (246{\cdot}4^\circ - 233{\cdot}1^\circ) = 16{\cdot}39.$$

The above results agree closely with the theoretical design calcu-ations, and indicate that the blading of the various expansions has ›een very accurately fitted

'otal B.T.U. for H.P. turbine per lb. steam

$$= 22{\cdot}78 + 22{\cdot}02 + 22{\cdot}95 + 23{\cdot}06 + 20{\cdot}79 + 17{\cdot}62 + 17{\cdot}51 + 16{\cdot}39 = 163{\cdot}12.$$

L.P. Turbines.

$$\text{Mean Entropy remaining} = \frac{1{\cdot}247 + 1{\cdot}515}{2} = 1{\cdot}381.$$

$$\text{Temperature Difference} = (233{\cdot}1^\circ - 102{\cdot}1^\circ) = 131.$$

Then,
$$\text{B.T.U.} = 1{\cdot}381 \times 131 = 180{\cdot}9.$$

Notice that the temperature and entropy at the terminal pressure position of the last H.P. expansion is taken, also the L.P. exhaust temperature and entropy, which is $\cdot 102{\cdot}1^\circ$ temperature and $1{\cdot}515$ entropy.

$$\text{Total B.T.U. per lb. steam} = 163{\cdot}12 + 180{\cdot}9 = 344{\cdot}02.$$

Assuming $13{\cdot}5$ lbs. steam per S.H.P. per hour,

Then,
$$\text{Turbine Efficiency} = \frac{33000 \times 60}{13{\cdot}5 \times 344 \times 778} = \cdot 548, \text{ or } 54{\cdot}8 \text{ per cent.}$$

R.—Manœuvring Trials of H.I.J.M.S. "Ibuki."*

Kinds of Trial.	Date.	Ahead.†		Astern.		$\frac{B}{A}$	Steam Pressure in Lbs.	
		Revolutions per Minute.	Shaft Horse-Power (A).	Revolutions per Minute.	Shaft Horse-Power (B).		Ahead.	Astern.
	1909.					p.c.		
Inertia	July 17	110	2200	80·6	1845	83·7	200·0	150·0
	„ 17	150	5000	78·9	2837	56·7	250·0	225·0
	„ 17	175	7800	94·7	3941	50·5	225·0	172·0
	„ 17	190	10200	98·5	4488	44·0	200·0	188·0
	„ 19	218	14750	109·6	5460	37·0	231·0	218·5
Astern	„ 19	200	10322	133·7	4291	41·6	232·5	210·0
	„ 19	120	2700	108·0	2585	95·5	224·0	200·0
	„ 19	85	1200	76·4	748	62·3	225·5	200·0
	Aug. 7	239	20612	196·8	12265	59·5	241·0	208·5

Notice.—In the manœuvring trial results the column marked $\frac{A}{B}$ gives the per cent. of stern power developed as compared to ahead power (100 per cent.) developed just before reversing.

* Reprinted from *Engineering*.

† The horse-power mentioned here is decided by the number of revolutions counted before thc order to go astern.

RESULT OF [illegible]

(CURTIS TURBINES.)

Date	Kinds of Trial	Duration of Trial in hours.	Revolution per Min. Mean	S.H.-P. Total	Number of Nozzles on Each Stage. Starboard. 1st	2nd	3rd	4th	5th	6th	7th	Port. 1st	2nd	3rd	4th	5th	6th	7th
1909. Aug. 12	Full power	6	250·2	27111	20	64	96	96	256	256	256	20	64	96	96	256	256	256
" 7	"	6	236·6	21004	16	64	96	96	256	256	256	16	64	96	96	256	256	256
July 31	"	24	216·7	16027	13	56	96	96	237	237	237	13	56	96	96	237	237	237
" 26	"	6	188·4	9812	9	64	96	96	256	256	256	9	64	96	96	256	256	256
" 24	"	8	152·2	5254	5	64	96	96	256	256	256	5	64	96	96	256	256	256

Date	Steam Pressure (in Pounds and Inches). Boilers.	Main Steam in Engine-Room. Starboard.	Main Steam in Engine-Room. Port.	Chest of Turbine. Starboard.	Chest of Turbine. Port.	1st Stage. Starboard.	1st Stage. Port.	2nd Stage. Starboard.	2nd Stage. Port.	3rd Stage. Starboard.	3rd Stage. Port.	4th Stage. Starboard.	4th Stage. Port.
1909. Aug. 12	260·6	252·1	251·4	244·4	238·9	71·40	67·20	39·70	40·00	12·00	12·40	−1·92	−0·88
" 7	261·8	246·3	247·3	242·0	236·6	54·50	49·40	29·00	29·30	+13·60	+13·60	−6·6	−6·88
July 31	253·5	244·3	239·8	234·2	232·8	47·90	41·90	30·10	31·00	+3·70	+6·10	−9·80	−9·00
" 26	260·4	239·7	235·2	231·9	228·2	23·50	19·10	8·90	9·40	−5·10	−6·20	−17·60	−16·70
" 24	244·9	227·2	229·4	220·0	219·6	8·50	7·20	−3·30	−0·60	−15·60	−14·60	−23·60	−23·00

(*Continued.*)

Date	Kinds of Trial	Steam Pressure (in Pounds and Inches). 5th Stage. Starboard.	5th Stage. Port.	6th Stage. Starboard.	6th Stage. Port.	7th St[age]. Starboard.	7th St[age]. Port.
1909. Aug. 12	Full power	−15·68	−15·41	−22·84	−22·55	−25·79	–
" 7	"	−18·10	18·40	−24·00	23·90	26·40	–
July 31	"	19·80	−19·80	−24·00	−24·80	−26·80	–
" 26	"	23·80	23·80	−26·80	−25·70	27·80	
" 24	"	26·00	25·00	27·70	27·00	28·10	–

Date	Temperature (Deg. Fahr.). Superheat after passing Regulating Valves. Starboard.	Superheat. Port.	Main Condenser. Sea Water. Inlet.	Main Condenser. Sea Water. Outlet.	Main Condenser. Condensed Water.	Feed Water after passing through Feed Heater.	Boiler. Number of Boilers. Double-Ended.	Number of Boilers. Single-Ended.	S.H.-P. per Square Foot of Grate.	Water Rate of Main Engine per S.H.-P. Not Corrected to Contract Conditions (per Hour).	Corrected to Contract Conditions (per Hour).	Main Condenser. Condensed Water per Sq. Ft. of Cooling Surface per Hour in Lbs.	Circulating Sea Water per Pound of Steam in Lbs.	Main Engine. Bucket Speed in Feet per Second.	Steam Velocity in Feet per Second (1st Stage).	Efficiency of Turbine. Per Cent.
1909. Aug. 12	21·8	22·9	74·6	97·7	90·0	131·11	10	8	16·45	15·050	18·77	11·506	44·296	157·3	2142·5	52·45
" 7	15·0	10·3	71·2	91·4	80·0	132·76	10	8	12·80	15·686	14·76	9·000	30·685	147·7	2330·0	48·95
July 31	6·8	3·9	75·0	93·4	88·5	151·6	10	8	9·72	16·506	15·62	7·253	41·118	136·2	2379·0	46·20
" 26	1·3	2·0	78·2	90·6	86·7	136·8	6	8	9·24	18·652	17·75	6·038	31·056	118·4	2650·0	40·70
" 24	3·1	4·3	78·2	92·3	88·7	145·2	5	0	9·05	21·074	20·57	8·065	30·946	95·6	2943·0	35·10

These tables of trial results give a fair idea of the pressures usually obtaining in the various expansion stages of large Curtis type impulse turbines, under varying conditions of speed and power.

In reading the results it should be noted that from the 3rd stage on to the 7th stage the plus and minus signs before the figures refer to inches of mercury and not lbs. of pressure. For example during the full power trials the 4th stage pressure is given as $-1{\cdot}92''$ starboard and $-0{\cdot}38''$ port, which means that the pressure at these stages was approximately 14·03 lbs. and 14·80 lbs. absolute, determined as follows:—

Then, $$\frac{29{\cdot}99'' - 1{\cdot}92''}{2} = 14{\cdot}03 \text{ lbs. absolute}$$

and $$\frac{29{\cdot}99'' - {\cdot}38''}{2} = 14{\cdot}80 \text{ lbs. absolute.}$$

At the 7th (last) stage during full power trials the reading was $-25{\cdot}79''$ starboard and $25{\cdot}64''$ port.

Then, $$\frac{29{\cdot}99'' - 25{\cdot}79''}{2} = 2{\cdot}10 \text{ lbs. absolute}$$

and $$\frac{29{\cdot}99'' - 25{\cdot}64''}{2} = 2{\cdot}17 \text{ lbs. absolute.}$$

Note.—**2″ Mercury = 1 lb. pressure.**
,, **Barometer, 29·99 inches.**

AT FULL POWER—

Steam Speed at 1st stage = 2142 ft. per second.
Blade Speed = 1137 ,, ,,
Turbine Efficiency = 52·45 per cent.

Variations in Turbine Design.

In marine practice the varieties of turbine design and turbine arrangement are numerous, and a few of the leading types are here detailed:—

1. Direct Drive Reaction Turbines with Two Shafts. (Parsons.)

This consists of one H.P. turbine, one L.P. turbine, and one reverse turbine: three turbines in all. The H.P. exhausts into the L.P., and the L.P. into the single condenser. The reverse turbine is contained in the L.P. casing, and takes direct boiler steam, the ex-

haust being led to the condenser. The turbine drums are parallel, and the casing stepped at each expansion.

The initial end of both H.P. and L.P. is forward, with the initial end of the reverse turbine aft.

2. **Direct Drive Reaction Turbines with Three Shafts.** (Parsons.)

In this arrangement five turbines are fitted—one H.P. on centre shaft, and two L.P. turbines are on each outboard L.P. shaft, the two reverse turbines being contained in the two L.P. turbine casings. The rotor drums are parallel, and the casings stepped as in the previous design. The initial end of both H.P. and L.P. ahead turbines are forward, and the initial ends of the two reverse turbines are aft. The L.P. turbines each exhaust into separate wing condensers, into which the reverse turbines also exhaust.

3. **Direct Drive Reaction Turbines with Three Shafts.** (Parsons.)

This arrangement consists of five turbines—one centre H.P., one wing I.P., and one wing L.P.; the reverse turbines, two in number, are fitted, one in the wing I.P. turbine casing, and the other in the wing L.P. turbine casing.

The turbine drums are parallel, and the casings stepped, the initial end of each ahead turbine being forward, and the initial end of each astern turbine aft. The L.P. turbine exhausts into the condenser when running ahead, and the reverse turbines exhaust to the condenser when running astern.

4. **Direct Drive Reaction Turbines with Four Shafts.** (Parsons).

In this design six turbines are fitted—one inboard H.P., one inboard I.P., and two wing L.P. turbines, in the casings of which are contained the two reverse turbines, or, in the case of large powers, the reverse turbines are arranged as separate units, and independent of the L.P. turbines.

The rotor drums are parallel, and the casings stepped, the initial end of all ahead turbines being forward, and the initial ends of the two reverse turbines aft.

The exhaust arrangements are similar to those of the previous designs.

5. **Direct Drive Reaction Turbines with Four Shafts.** (Parsons.)

In this arrangement eight turbines are fitted in all—four ahead and four astern.

The two ahead H.P. turbines are fitted inboard, and the two ahead L.P. turbines outboard, while one reverse turbine is fitted on each shaft as an independent unit.

The initial ends of both H.P. and L.P. turbines are forward, and the initial ends of the reverse turbines may either be forward or aft as found most suitable, this alternative being permissible when the astern turbines are independent of the L.P. ahead turbines.

As before, the rotor drums are parallel and the casings stepped.

The exhausts of ahead and astern turbines are similar to those of previous designs, two wing condensers being fitted.

6. Direct Drive Combined Impulse and Reaction Turbines with Two Shafts. (Parsons.)

In this arrangement only two ahead turbines and two astern turbines are fitted, or four in all, the astern turbines being contained in the same casings as the ahead. The ahead turbines are of the "complete expansion" type, as the steam expands direct from stop valve pressure to condenser in the one turbine, which, therefore, forms a combination of both H.P. and L.P.

The initial stage of both ahead and astern consists of an impulse wheel, which is followed by a parallel drum for the reaction stages. The initial end of each ahead turbine is forward, and of each reverse turbine aft and two wing condensers are fitted, into which the ahead turbines exhaust when running ahead, and the reverse turbines when running astern.

7. Direct Drive Impulse Turbines with Two Shafts. (Curtis.)

In this design two ahead turbines and two reverse turbines are fitted, or four turbines in all, the reverse turbines being contained in the ahead casings, each shaft having one ahead and one astern turbine connected to it.

The ahead turbines are, like design No. 6, of the complete expansion type, the steam expanding direct from stop valve pressure to condenser in each single turbine casing.

The ahead turbines consist of from five to seven ahead pressure impulse wheels keyed to the shaft, each wheel being compounded for velocity by having two or three moving blade rows or stages.

The astern turbines are fitted with two or three pressure stage-wheels, each wheel being again compounded for velocity by the fitting of two blade rows.

The initial ends of the ahead turbines are forward, and of the astern turbines aft.

Two wing condensers are fitted, into which the turbines exhaust when running either ahead or astern.

8. Direct Drive Impulse Turbines with Two Shafts. (Curtis.)

In this arrangement (Brown-Curtis) the turbines are again four in number, and are arranged similar to the last described, but in this

case the L.P. ahead turbines are fitted at the initial stages with from three to five impulse wheels, followed by from six to ten additional impulse stages placed on a tapered or stepped-down drum in place of on separate wheels. A further and later date modification of this is in the substitution of reaction blading stages on the tapered drum, which are then increased in number (face p. 434) and may run to from twelve to twenty stages or, as commonly expressed, "expansions." The blading of the drum section is similar to the Parson's blading. The astern turbines usually consist of impulse wheels only, compounded for velocity by having two or more blade rows.

The ahead wheel stages are also compounded for velocity in the same manner.

9. Geared Down Turbines with Two Shafts. (Parsons.)

In this arrangement six turbines are fitted in all—one H.P. ahead, one L.P. ahead, and one astern to each shaft.

The H.P. ahead turbines are occasionally fitted with an initial impulse stage wheel, followed by the usual reaction blading in a stepped drum, but in some cases the impulse stage is omitted.

The L.P. ahead turbines consist of reaction blading only, while the astern turbines have an initial impulse wheel followed by a few rows of reaction blading. The impulse wheels of both H.P. and astern are compounded for velocity by having two or three moving blade rows fitted. The initial end of the H.P. turbines is forward, the initial end of the L.P. ahead turbines aft, and the initial end of the astern turbines forward (face p. 480). To take up the unbalanced steam pressure on the motor blade rows, balance pipes and double dummies are fitted to the H.P. turbines.

10. Geared Down Turbines with Two Shafts. (Curtis.)

In this modern arrangement three turbines are connected by gear pinions to each shaft—one H.P., one L.P., and one astern, or six turbines in all, the astern turbines being contained in the L.P. ahead turbine casings.

The H.P. turbines consist of from five to six impulse wheels and stage diaphragms, the initial stage wheel only being compounded for velocity by having two or three blade rows, the other stages having only a single blade row. From this it will be evident that one row only of guide blades will be necessary in the casing at the first stage. The L.P. turbines also consist of a series (six or seven) impulse wheel stages with single blade rows on each.

The astern turbines usually consist of a single impulse wheel fitted with three velocity blade rows (moving) and two rows of guide blades (fixed), being thus compounded for velocity only and not for pressure.

The initial ends of the H.P. turbines are forward, and of the L.P. turbines aft, while the initial ends of the reverse turbines are forward.

Nozzle control valves are fitted to the H.P. ahead and to the astern turbines, but, as will be evident on consideration, these require to be omitted in the case of L.P. turbines or intermediate turbines.

A modification of the foregoing consists in the fitting of two reverse turbines to each shaft, one H.P. reverse turbine placed forward of the H.P. ahead, and one L.P. reverse turbine placed forward of the L.P. ahead, and as usually arranged. As in previous designs, each set of turbines (port or starboard) is supplied with its own condenser and pumps, etc.

General Notes.

The turbine revolutions for direct drives vary from 160 to 400 per min., the slower speed being for curtis velocity compounded turbines, and the higher for pure reaction turbines of the Parsons class. The turbine revolutions for geared down drives range from 1500 to 3000 or 3500 per min., the higher speeds referring to double reduction gearing. The gear down ratio may be anything from 6 to 16 for single reduction, and 30 to 50 for double reduction gearing.

The turbines a observed from the photo, the H.P. turbine (furthe on the left, also steam strainer above. The L.P. le rows. The astern turbine is supplied with an i control valves fitted. The gear wheel case (on righ low the vapour formed from oil at fairly high tempe is fitted with a vertical joint in addition to the usu spindle and pinion, oil pressure pumps, etc., etc.

SECTION XII.

TURBO-GENERATORS.

Principal Differences between Marine Turbines and Turbo-Generators. (Turbines driving Dynamos.)

1. Turbines for driving electric generators (light or power) are, in most cases, of the impulse type.

2. Are run at higher revolution speed than marine turbines for either direct drive or gear down drive.

3. Are fitted with automatic governors (blast or relay types) to keep revolution speed constant within about 5 per cent. under or over normal.

4. Are also fitted with emergency governors which act to instantly shut off the steam in case of accident causing sudden release of load.

5. Are often fitted with spherical seated bearings to allow of flexibility.

6. Are often fitted with special water-sealed glands in addition to labyrinth packing rings of the usual type.

7. Have usually a flexible claw type coupling between the turbine and the dynamo to allow for turbine adjustment, and for heat expansion.

8. Lubrication of the bearings, if not of the forced pressure system is of the oil ring system.

9. The coal and water consumption is less per H.P. hour (or as it is usually measured, per K.W. hour) than in the case of marine turbines.

Note.—Volts × Amps. = Watts, and 1000 Watts = 1 Kilowatt, also, 746 Watts = 1 H.P.
Then, 1000 ÷ 746 = 1·34 H.P. per Kilowatt.

In general turbine practice turbo-generators give the best efficiency results, and are, in most cases, superior in this respect to marine turbines.

10. The use of superheated steam is more general than in marine practice.

Types of Turbo-Generators.

No. 1.—Sulzer Type Turbine.

This turbine is arranged with a three-blade row velocity impulse wheel at the first stage, and Parsons reaction blading for the following stages, the drum and casing being stepped. Dummy pistons are omitted in this design, the steam thrust being balanced by a special oil piston (shown on the left) which is connected to the small thrust bearing. On the right is shown the flexible coupling connecting the turbine to the generator, and on the

No. 2.—Brown-Boveri Turbine.

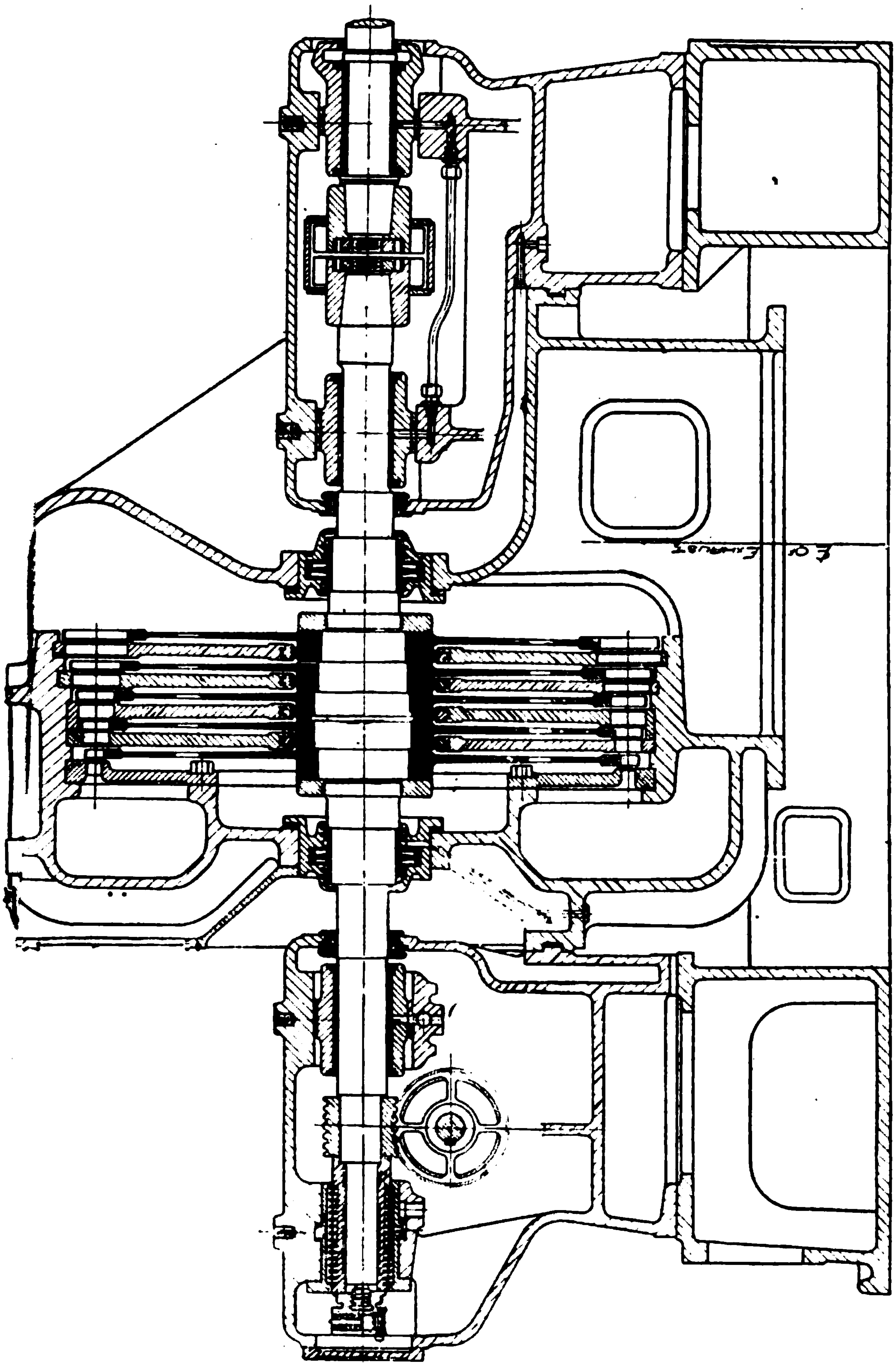
This type is fitted with an impulse wheel at the first stage having two velocity blade rows or stages, the expansions which follow consisting of Parsons reaction blading. The drum and casing are both stepped, and drainage openings are shown between the steps of the drum. Three main bearings are fitted, and this feature, it should be noted, is special to turbines of the electric generator class. A flexible coupling aft and a thrust forward are fitted as shown, and the governor and oil pump are driven by the worm shown on the end of the turbine spindle (left); the spring loaded governor gear is also fitted on the turbine in front of the impulse stage nozzle opening.

No. 3.—Zeolly Turbine.

The type shown consists of a number of impulse pressure stages (12), each with one velocity blade row.

The casing is fitted with nozzle diaphragms only between each stage, no blade rows being necessary.

Three bearings are fitted, and an adjusting block and governor gear are placed at the forward end.



No. 4.—British Westinghouse Low Pressure Turbine.

The turbine illustrated is supplied with five impulse pressure stages of the Rateau type, has three bearings, a flexible coupling, a thrust, and worm gear for oil pump and governor control. Each impulse wheel stage shown is arranged with full peripheral admission.

No. 5.—Back Pressure Turbo-generator.

(The British Westinghouse Co. Ltd.)

The drawing shown above represents a low speed ship-lighting set of 400 Kilowatt output, running at 1500 revs. per min., and driving a direct current generator exhausting against a back pressure of 20 lbs. absolute (5 lbs. gauge). The turbine consists of three pressure stages, each formed of two velocity blade rows, or stages, and is therefore compounded for both pressure and velocity. It will therefore be evident that each diaphragm plate contains one row of fixed guide blades. A flexible [illegible] shown on the right, and the usual rotor adjusting block and worm drive on the left.

The British Thomson-Houston Co. Ltd., Rugby, own the patent rights of the Curtis Steam Turbine which they have manufactured at their Rugby works for some considerable time, and which they apply to turbo-generators.

By the courtesy of the British Thomson-Houston Co. Ltd. the author is enabled to reproduce the following notes, descriptions, and illustrations of turbo-generators.

Impulse Turbine.

An impulse turbine is distinguished by the fact that the potential energy of the steam is transformed into kinetic energy by expansion of the steam through specially shaped stationary nozzles. As a

No. 6.—Diagram of Simple Impulse Turbine.

result of this the steam acquires great velocity, whereupon it is brought in contact with suitable blades or buckets mounted on the periphery of a wheel or wheels free to rotate, the buckets being shaped so as to turn the issuing jet of steam gradually and without shock in a backward direction. This is shown diagrammatically in No. 6.

The impact of the steam on the buckets results in the steam giving up the whole or part of its velocity and consequent energy to the buckets, thus causing rotation of the wheel.

It is sometimes argued that the velocity of the steam in an impulse turbine is liable to cause damage to the moving buckets through erosion, but this has been proved not necessarily to be the case, as the buckets of Curtis turbines which have been running for

some years show on examination only the very slightest traces of wear.

Principle of Impulse Turbine.

A mass which has been moving at a certain velocity will, in coming to rest, give up the same amount of energy as was required to give it the velocity it previously possessed. The object to be attained in the design of an impulse turbine, therefore, is to arrange the buckets so that they will bring the jets of steam issuing from the nozzles to as near rest as possible, that is to say, that the steam, after passing through the buckets of the revolving wheel, shall possess no motion and consequently no kinetic energy.

Best Bucket Speed.

It is easy to see that if a bucket is moving as fast as the steam jet it will offer no opposition to the jet, nor will it extract any of the velocity.

If, on the other hand, the wheel is so secured as to be immovable, the steam jet will be directed backwards and will rebound from the bucket with the same velocity with which it entered it, neglecting friction in the bucket.

If, for the sake of example, however, the jet is assumed to be travelling at the rate of 1000 ft. and the bucket at 500 ft. per sec., then the steam will strike the bucket with a relative velocity of 500 ft. per sec., and on issuing from the bucket will be directed backwards with a velocity relatively to the bucket of 500 ft. per sec.

Since, however, the bucket is moving forward at the rate of 500 ft. per sec., the backward velocity of the jet is equal and opposite to the forward velocity of the bucket, so that the resultant velocity of the steam relatively to a fixed point in space, as it emerges from the bucket, is zero. The emerging steam accordingly is inert, having given up all its kinetic energy to the bucket.

It follows from this that for the best efficiency the buckets should move at half the velocity of the steam jet.

As a matter of fact, steam when expanded from 150 lbs. pressure per sq. in. to atmospheric pressure attains a speed of 2950 ft. per sec., while, if expanded into a 28 in. vacuum, it can attain a velocity of 4010 ft. per sec., or nearly twice the speed of a modern rifle bullet, so that it is evidently impracticable to construct a wheel which can run at half the velocity of such a jet.

Curtis Turbine.

The Curtis turbine, while utilising the principle of expanding the steam through specially-shaped nozzles, has the great advantage that it can be run with high efficiency at a relatively low speed.

This is accomplished in two ways, by means of—(1) "velocity stages," and (2) "pressure stages."

Velocity Stages.

If a turbine such as just considered be run comparatively slowly, the buckets on the rotating wheel will absorb less of the velocity of the steam jet, but the steam emerging from the buckets will still have considerable velocity, and will be available for use over again.

No. 7.—Arrangement of Moving and Stationary Elements in a Curtis Turbine.

If, therefore, the steam on emerging is redirected to a second row of moving buckets fixed to the same wheel as the first row, it may be compelled to part with its remaining velocity in the second row of moving buckets.

This method of fractionally extracting the velocity of the steam, and known as a "velocity stage," is used to a differing extent in the design of both compound impulse, and combined impulse, Curtis turbines.

Steam entering the diverging nozzles (No. 7) at working pressure is expanded during its passage through them, and issuing at great velocity enters the first row of moving buckets. The steam leaves these buckets in a backward direction and enters the ring of stationary

buckets, in which it has its direction reversed, so that it enters the second ring of moving buckets in the same direction as at first.

The nozzles are designed so that practically all the potential energy of the steam is converted into kinetic energy in the jet. In other words, the steam in passing through the nozzles is expanded down to the pressure of the chamber or stage in which the wheels are revolving, so that after leaving the orifice of the nozzle there is practically no tendency for any further expansion, the steam acting on the buckets in that stage simply by impact.

As a result of this action there is no appreciable difference of pressure between the points of entrance and exit of the various rows of buckets in a stage, and consequently no end thrust nor tendency for the steam to leak across the clearance space, so that the moving buckets of the Curtis turbine need no special minimum clearance from the casing

Pressure Stages.

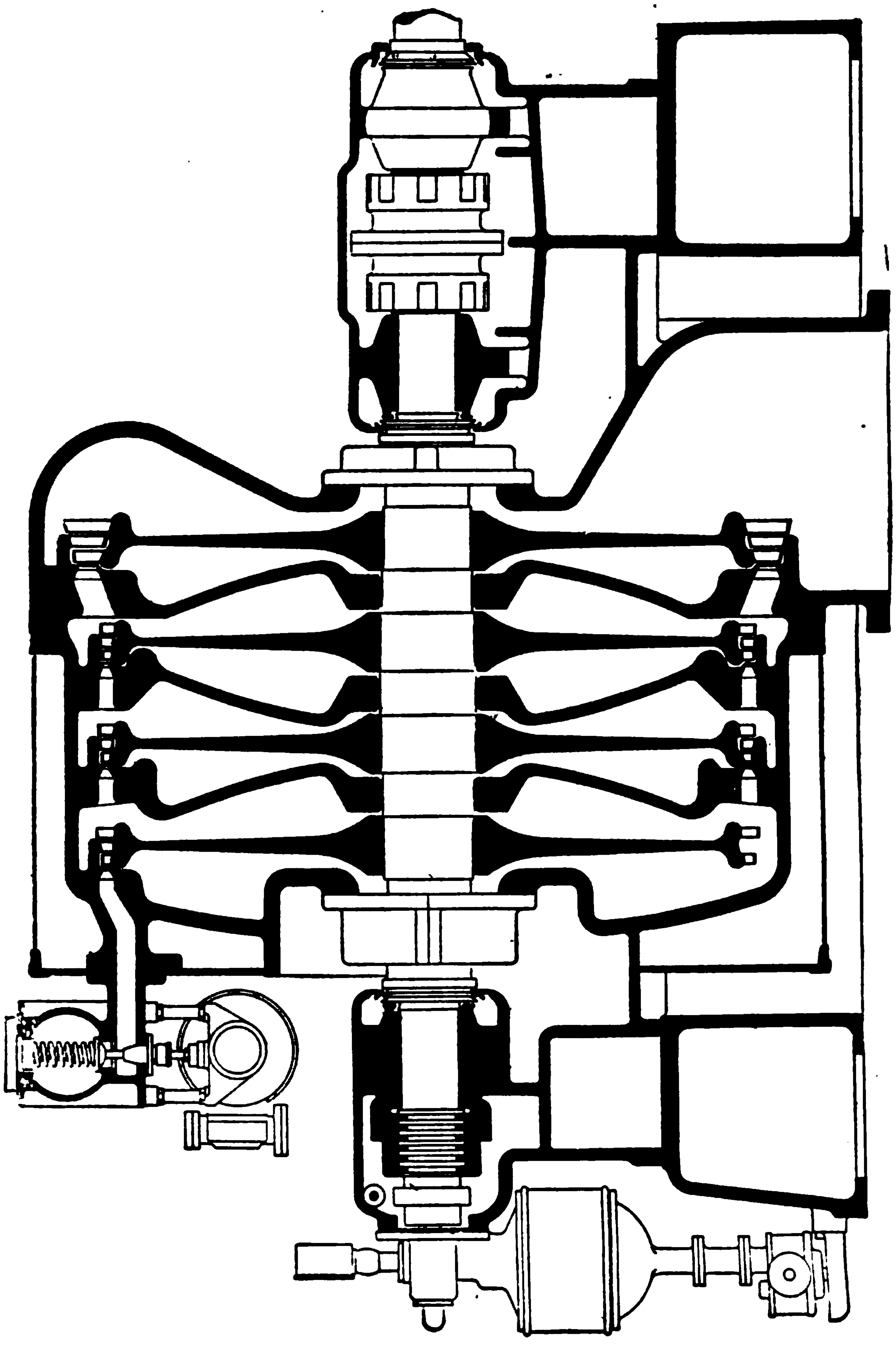
It is obvious from the description just given that any degree of expansion, with its resulting steam velocity, can be dealt with in one pressure stage by providing a sufficient number of rows of buckets or "velocity stages" to bring the speed of rotation down to a practicable limit.

This method, however, if pushed to an extreme, becomes inefficient due to steam friction losses, and so another method of subdividing the steam energy is also utilised in the Curtis turbine. Instead of expanding the steam from boiler to exhaust pressure in one step, this operation is divided over two or more sections or "pressure stages."

The multi-stage machine can be best understood by imagining a complete turbine to consist of a number of smaller turbines placed in series. The distribution of steam pressure is regulated by the size of the exhaust opening in each section, which opening usually forms the nozzles for the succeeding section. No. 8 represents a four-stage turbine, and clearly shows how the machine is divided up into four sections or stages by means of steam-tight diaphragms, which also carry the nozzles.

Each set of nozzles is designed to utilise one quarter of the total energy in the steam during expansion from the boiler down to the exhaust pressure. This being so, a relatively low velocity is imparted to the steam jet in each stage, which in turn permits of a comparatively low speed of rotation of the wheels.

The steam, after leaving the last row of moving buckets in the first stage, has had nearly all of its velocity extracted, but on passing through the nozzles in the diaphragm separating the first and second stages it is again expanded and the same velocity as before imparted to it. This occurs again in the third and fourth stages, the steam



No. 8.—Longitudinal Section of 4-Stage Curtis Compound Impulse Turbine.

finally emerging from the latter at the pressure of the exhaust, having given up practically all the energy it previously possessed when in the boiler. It will be noted that the nozzles and buckets of the successive stages have to be made gradually larger to deal with the increase in volume of steam resulting from each expansion.

The simplest form of multi-stage machine is found in the simple impulse turbine, which consists of a number of pressure stages having but one row of moving buckets in each stage.

Curtis Compound Impulse and Combined Impulse Turbines.

To obtain the highest efficiency necessitates the proper blending of "velocity steps" (number of rows of buckets per stage) with "pressure steps" (number of stages). These conditions have received the most careful consideration in the design of the Curtis machine, hence the adoption by the British Thomson-Houston Co. of the two main types of turbines, viz., the compound impulse and the combined impulse machines. The choice of type employed in any instance is determined by the conditions to be met.

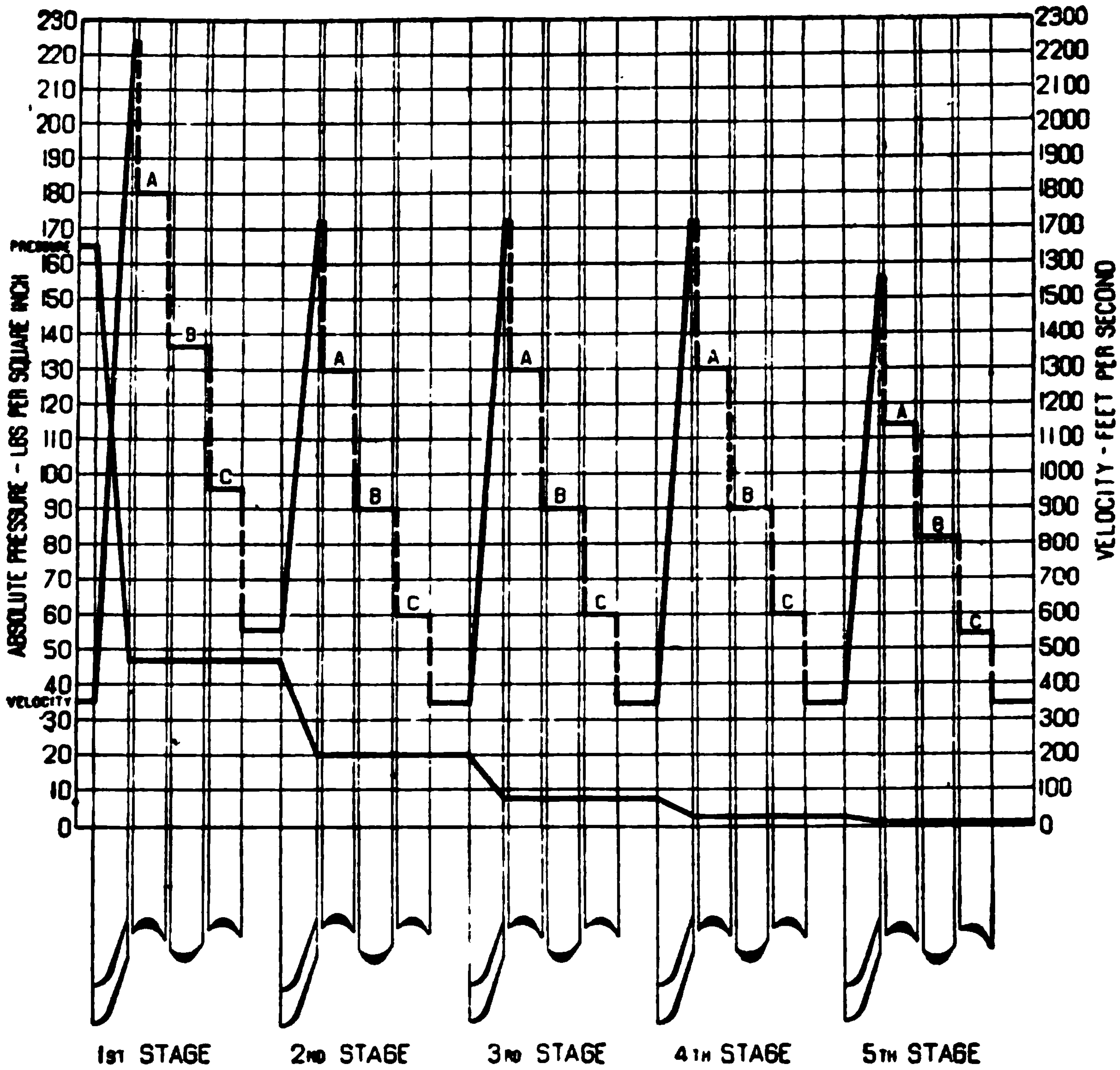
The difference between the two lies in the fact that in the compound impulse turbine there are two or more rows of moving buckets in each stage of the machine, while in the combined impulse turbine there are two rows of moving buckets in the first stage, but only one row in each of the following stages.

The compound impulse turbine, as a consequence, is a type possessing few stages and is very short in length. The combined impulse turbine, requiring as it does a greater number of stages than the compound impulse machine, is longer over all. In the larger sizes, under certain steam conditions, the combined impulse turbine offers better economy than the compound impulse machine.

Nos. 9 and 10 show the steam pressure and velocity curves of a typical 5-stage high pressure compound impulse and an 11-stage combined impulse turbine respectively. These show at a glance how the pressure drop through the different nozzles varies so as to obtain, as far as possible, the liberation of the same amount of energy in each stage, the latter being evidenced by the height of the velocity curve at each expansion.

In the compound impulse turbine (No. 9) it will be seen that in the first stage the pressure drop is much greater in proportion to the kinetic energy developed than in the second stage, whilst in the third and fourth stages the pressure drop decreases, although producing the same kinetic energy as in the second stage. In the fifth stage, whilst the energy liberated is nearly as great as for the three previous stages, the pressure drop here is insignificant. This shows indirectly how an increase in vacuum (*i.e.*, a drop in back pressure) adds so rapidly to the amount of energy available per lb. of steam.

In the combined impulse turbine (No. 10) the pressure drop in the first stage is also greater in proportion to the kinetic energy liberated than in the succeeding stages; but the kinetic energy liberated in the first stage is very much greater than in the stages which follow. In fact this is the peculiarity of the combined impulse turbine, and under certain conditions it is conducive to economy, a point which will be referred to later.



No. 9.—Steam Pressure and Velocity Curves of 5-Stage Curtis High Pressure Compound Impulse Turbine.

Referring to the velocity curve (No. 9) the line A shows the steam velocity through the first row of moving buckets, while B and C show the velocity through the intermediates and second row of moving buckets respectively. In No. 10 the lines A, B, and C occur in the first stage. In the succeeding stages, only the line A occurs, as there are no intermediates nor second row of moving buckets, and it will be seen that the velocity A in these stages is about the same as velocity C in the first stage.

It will be noticed in both Nos. 9 and 10 that the velocity through the first row of buckets is much less than the velocity at the nozzle

No. 10.—Temperature, Steam Pressure, and Velocity Curves of 11-Stage Curtis High Pressure Combined Impulse Turbine.

exit; the reason for this, as already explained, is that the buckets are moving with considerable velocity in the same direction as the steam, so that the relative velocity of the steam as it enters the buckets is

very much less than that with which it leaves the nozzles. The same effect occurs in the intermediates, and again in the second row of moving buckets, where these exist.

It will be seen that where two rows of moving buckets are employed, the steam velocity through the intermediates (*i.e.*, the line B) is approximately one-half of the velocity at the nozzle exit, and as the energy varies as the square of the velocity, this shows that after passing through the first row of buckets the steam has only 25 per cent. of the kinetic energy it possessed when leaving the nozzles. The first row of buckets, therefore, extracts about 75 per cent. of the energy available per stage, and the last row, which extracts the remaining velocity of the steam, accounts for the remaining 25 per cent. of available energy.

From this it can be reasoned that a stage having a single row of moving buckets is in general only able to deal with 25 per cent. of the energy which a stage consisting of two rows of moving buckets can absorb.

It is sometimes contended that the bending stresses in the buckets due to the impulsion of the steam are considerable in the compound impulse turbine, especially at the low pressure end where the buckets are long. This, however, is not the case. The buckets of the first row in each stage are thick at the centre on account of their small inlet and outlet angles, also comparatively short, so that although 75 per cent. of the energy per stage is extracted here, the bending stress is very low.

The second row of moving buckets is much thinner in section and somewhat longer, but owing to the small amount of kinetic energy extracted, the bending stress is again very low.

Provision for Decreased Velocity of Steam.

It will be seen in Nos. 8 and 16 that the successive rows of buckets in any given stage differ in length one from another. No. 11 shows diagrammatically and on a larger scale the relation of moving and stationary buckets in any one stage of a Curtis compound impulse turbine. This figure also represents the first stage of a combined impulse turbine, while the succeeding stages are as shown in No. 12. Referring to No. 11, A and C are the moving and B the stationary lines of buckets respectively. It might be assumed that the divergence of the bucket area as shown is to provide for expansion, and consequently increasing volume per unit weight of steam. This, however, is not exactly the case.

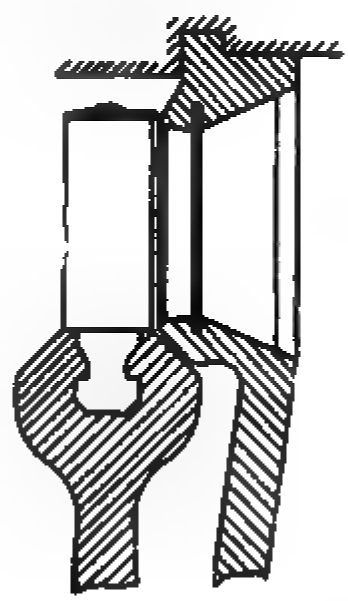
Steam enters A at a higher velocity than it leaves C. During its passage a continuous fall in velocity takes place, so that an increasingly wider passage and consequently longer buckets are necessary to accommodate the steam as it gradually moves more slowly.

This action may be compared with the flow of a non-expansive fluid such as water through a funnel-shaped tube. If the tube be

filled with water moving from the smaller to the larger end, the velocity will be high at the entrance, but relatively low at the discharge end. Thus the velocity of the liquid drops considerably but without expansion or increase in volume.

Some Advantages of B.T.H.-Curtis Turbines.

A feature of the greatest importance in connection with the Curtis, as compared with the reaction turbine, is that the buckets at the inlet end are not subjected to the effects of steam at high temperature. In expanding through the first set of nozzles the steam temperature falls considerably, so that no risk of damage to the buckets arises, even when using steam superheated to the highest degree.



No. 11 and No. 12.—Relation of Fixed and Moving Buckets in Curtis Turbines.

One of the most important advantages of the Curtis turbine is that in the event of a quantity of water coming over from the boilers with the steam, there is little danger of stripping the buckets or other damage.

Curtis turbines can be started up promptly from a cold condition without danger from accumulation of condensed steam, or fear of distortion due to change of temperature of the casing.

As compared with the reaction turbine, the Curtis machine shows to great advantage in the matter of temperature fluctuation.

It is not always possible to keep the steam pressure and superheat constant, and when these fluctuations occur they are, in the case of reaction machines, transmitted direct to the turbine.

These changes, however, are of small importance in the Curtis turbine, owing to the expansion of the steam in the first set of nozzles, in which the fluctuations of pressure and temperature are damped down before the steam enters the turbine proper. Thus,

there is not the same tendency for mechanical distortion in the Curtis as is present in the reaction machine.

In the combined impulse turbine, the first wheel with a double row of moving buckets takes the place of the first three wheels in a simple impulse turbine, and the advantages gained are as follows:—

(1) The density of the steam surrounding the single compound wheel is about two-thirds of the mean density of the steam surrounding the three single-row wheels, causing the compound wheel to have only about one-third the rotation losses of the latter wheels.

(2) The lower stage pressure associated with the single compound wheel causes less leakage of steam through the diaphragm and end packing. This leakage is further reduced by the fact that, as compared with the simple impulse machine, the Curtis turbine requires fewer stages, and so is shorter in length. This enables a smaller shaft diameter to be used than is possible with a longer machine, and so offers a considerably reduced area for steam to leak through the diaphragm packing.

(3) It is possible in the combined impulse turbine to obtain overloads without by-passing the first stage. In a simple impulse turbine, unless the energy drop through the first wheel is very large, it is not possible to obtain overloads without by-passing, so that whether the method adopted is to allow an abnormal energy drop or to by-pass the steam, the result is loss of efficiency.

The design of the Curtis turbine allows ample room for substantial wheel bosses, as well as for efficient diaphragm packings. A sound mechanical construction is, therefore, possible—(1) entirely obviating danger from the bucket wheels opening at the boss and becoming slack upon the shaft due to centrifugal force, and (2) effectually preventing leakage of steam from one stage to another.

Curtis High Pressure Steam Turbines.

Steam turbines are peculiarly suited for working not only at the highest pressure, but with a superheat and vacuum quite outside the reach of the reciprocating steam engine.

Under these conditions greatly increased economy is obtained by the steam turbine, and as the principles on which the conditions may be profitably utilised are of great importance in connection with any steam turbine proposition, they are clearly outlined in the examples which follow.

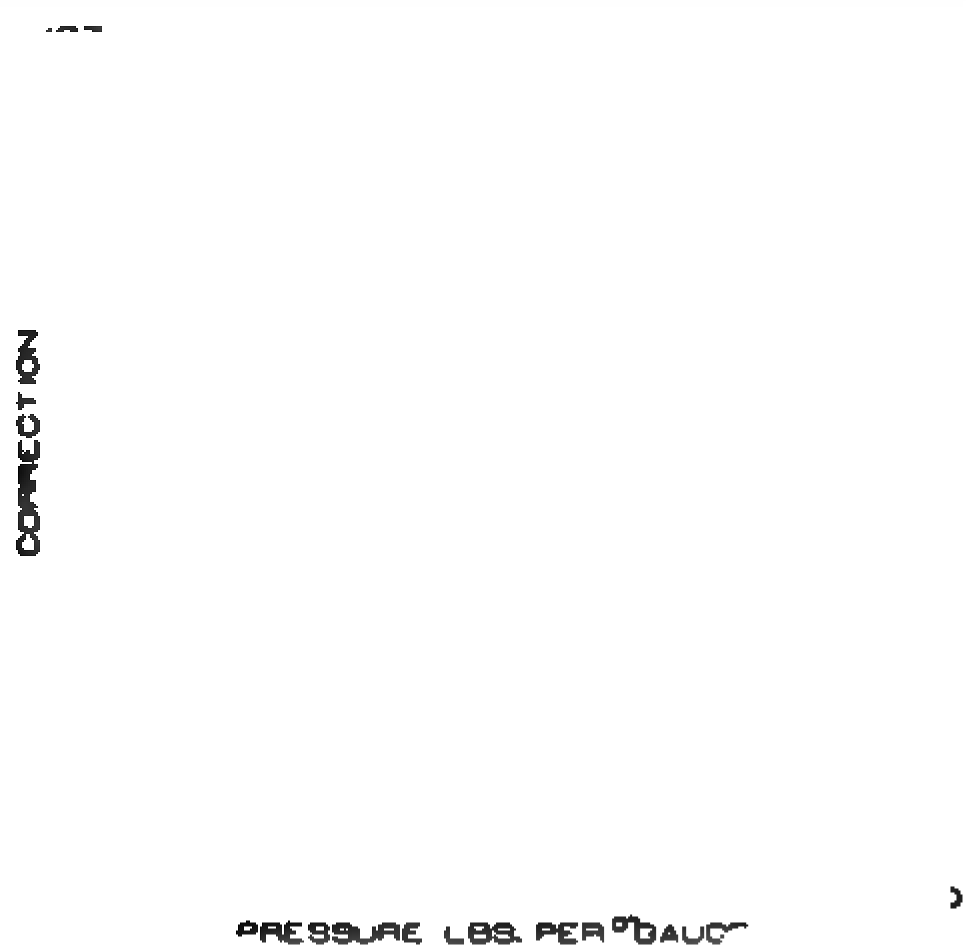
Gains due to High Pressure, Superheat, and Vacuum in the Curtis Turbine.

The curves shown in Nos. 13, 14, and 15, as well as the following examples derived from actual tests taken with Curtis turbines,

illustrate the substantial gains in economy which are realised by the use of high pressure, superheat, and vacuum.

Increase of Pressure.—Example I.

If, with dry steam at 150 lbs. per sq. in. gauge and 28 in vacuum, the steam consumption be equal to 1·0, at 200 lbs. per sq. in. gauge pressure it becomes on an average equal to 0·96 (No. 13), *i.e.*, a decrease in steam consumption of 4 per cent. Taking for convenience theoretical figures, and assuming that the condensed water from the



No. 13 (Example I.).—Correction Curve for Steam Pressure with 28 in. Vacuum.

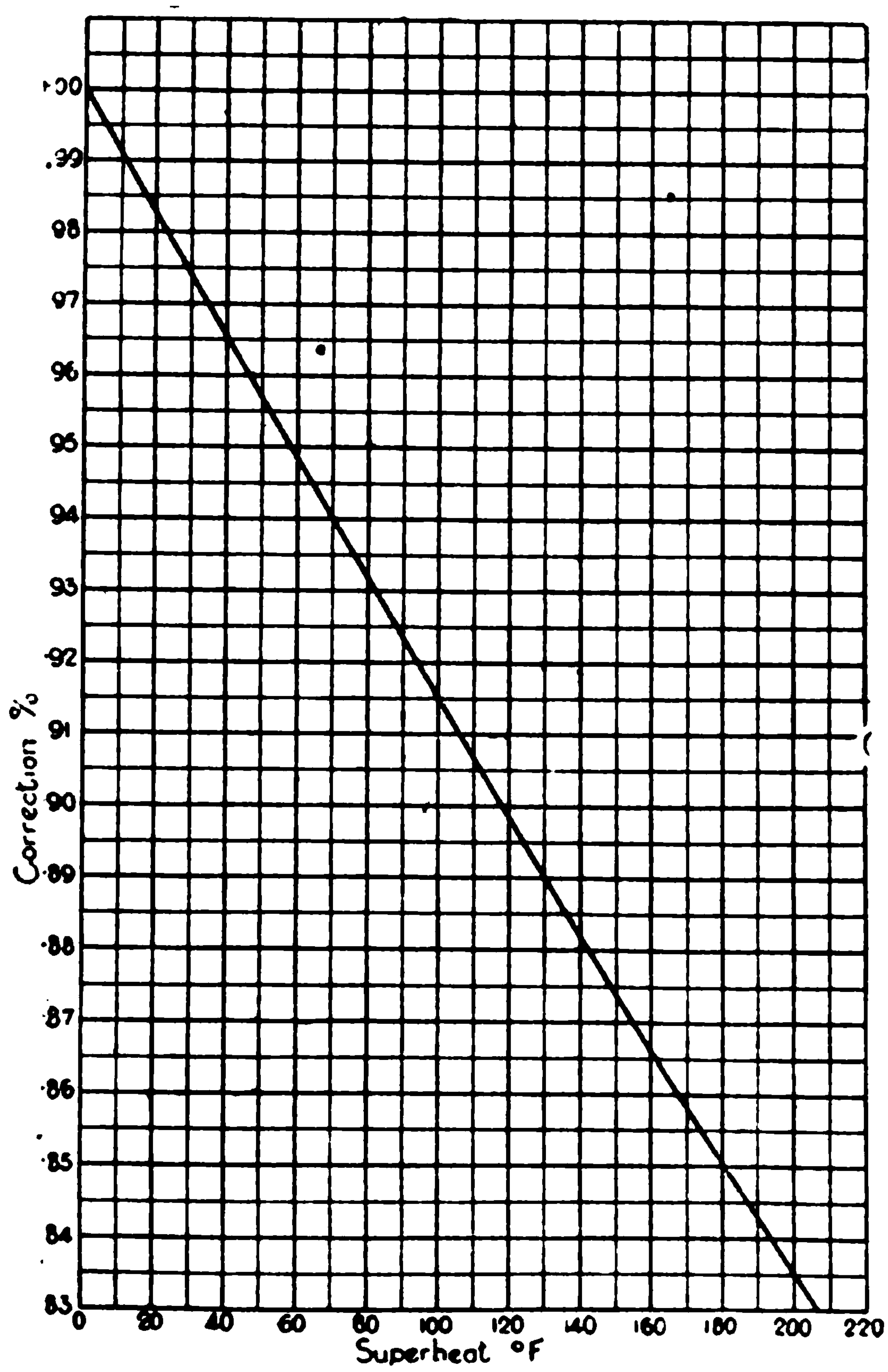
High Pressure Curtis Turbines.

condenser is used as boiler feed, and that its temperature is equivalent to that for a vacuum of 28 in., viz., 101° F., it will require 1125·7 units of heat to evaporate 1 lb. of this feed and raise it to a condition of dry saturated steam at 150 lbs. per sq. in. gauge pressure. In order to raise the same feed water to dry saturated steam at 200 lbs. gauge pressure it will require 1129·9 units of heat, a difference of only 4·2 British thermal units, *i.e.*, 0·37 per cent. increase.

By means, therefore, of an increase of 0.37 per cent. in the number of heat units put into the steam (by increasing its pressure from 150 lbs. to 200 lbs. per sq. in. gauge), the steam consumption is reduced by 4 per cent.

Increase of Superheat.—Example II.

If, with dry steam at 150 lbs. per sq. in. gauge and 28 in. vacuum, the steam consumption be equal to 1·0, at 100° F. superheat it becomes on an average equal to 0·915 (No. 14), thus showing a decrease in steam consumption of 8·5 per cent.



No. 14 (Example II.).—Correction Curve for Superheat.

High Pressure Curtis Turbines.

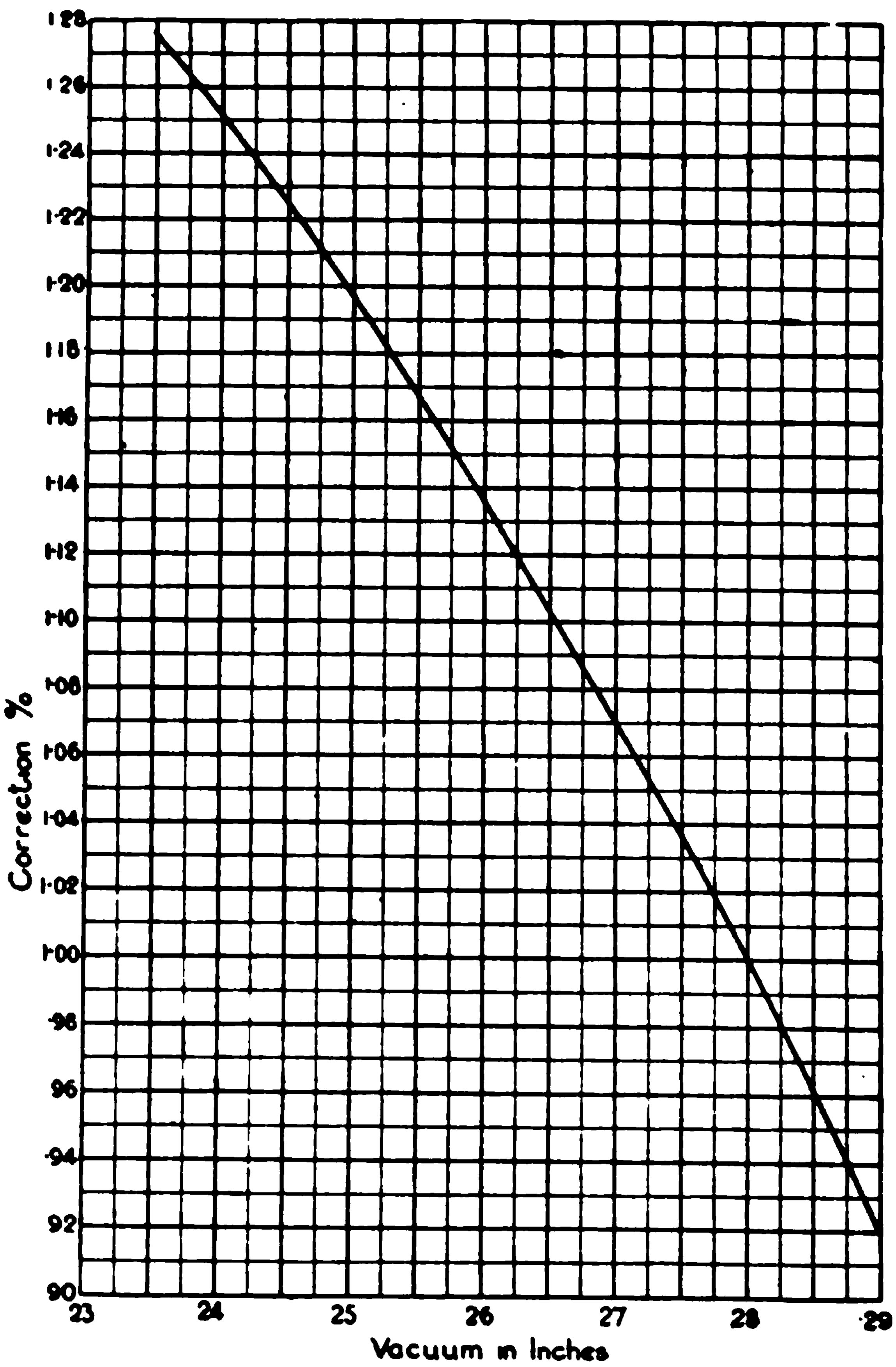
As stated in Example I., it requires 1125·7 units of heat to evaporate 1 lb. of the boiler feed to dry saturated steam at 150 lbs. per sq. in. gauge pressure. In order to raise the same feed to the same pressure with 100° F. superheat it requires 1182·9 units of heat, a difference of 57·2 B.Th.U.

By means, therefore, of an increase of 5 per cent. in the number of heat units put into the steam (by superheating steam at 150 lbs.

per sq. in. gauge, 100° F.), the steam consumption is reduced by 8·5 per cent.

Increase of Vacuum.—Example III.

If, with dry steam at 150 lbs. per sq. in. gauge and 28 in.



No. 15 (Example III.).—Correction Curve for Vacuum.
High Pressure Curtis Turbines.

vacuum, the steam consumption be equal to 1·0, under the same conditions, but with a vacuum of 28½ in., it becomes on an average equal to 0·96 (No. 15); that is to say, the steam consumption is increased by 4 per cent.

As indicated before, it requires 1125·7 units of heat to evaporate 1 lb. of the boiler feed to a condition of dry saturated steam at 150 lbs. per sq. in. gauge. In order to raise the feed water from a

temperature corresponding to a 28½ in. vacuum, *i.e.*, 92° F., it will require an additional 9 B.Th.U. per lb.

For an increase of 0·8 per cent., therefore, in the number of heat units put into the steam (due to the lower feed temperature) a reduction of 4 per cent. is effected in the steam consumption.

The gain mentioned is on account of the increased energy available due to the greater range of temperature through the turbine. Whether or not an actual net gain in economy will be effected by an increase of vacuum depends on the extra amount of power required for the condenser pumps, and upon the temperature of the cooling water available. This question must of necessity always be settled by due consideration of the conditions applying to individual cases. As regards the turbine itself, however, the gain due to increase of vacuum is always realised, as illustrated in Example III.

Construction.

In No. 16 is shown a longitudinal section of a large high pressure 11-stage Curtis combined impulse turbine. The steam, passing through the valve chest shown at the top left-hand side, proceeds down the passage to the nozzle plate, and, expanding through the nozzles, enters the moving and fixed rows of buckets of the first stage. This stage comprises a single wheel having two rows of buckets with a row of fixed intermediate blades between. From the first stage the steam passes through the expanding nozzles in the dish-shaped diaphragm and enters the buckets of the second stage. From the second stage the steam passes to the nozzles and buckets of the third stage, and so on through the succeeding stages, finally exhausting through the opening at the bottom right-hand side of the casing. The second and successive stages all contain wheels with a single row of buckets.

A great advantage of the Curtis turbine lies in the fact that, after expansion through the nozzles at the high pressure or governing end of the turbine adjacent to the main bearing, the temperature and the pressure of the steam are very much reduced, as will be seen on referring back to No. 10.

As a result the steam sealing glands, the end cover, and the main bearing at the high pressure end of the turbine are not subject to the excessive temperatures and pressures found in turbines of other types.

Casing.

The casing is made of high grade cast iron, and serves to hold the rows of fixed buckets and also the steam-tight diaphragms separating the stages. Due to the principle of the Curtis turbine just mentioned, the pressure of the steam is reduced in the first stage nozzles, before entering the casing, to a greater extent than in any other type of turbine, and as a consequence the internal pressure in the first stage of even a many-stage machine is not greater than about 25 per cent. of the boiler pressure under full load conditions.

In the B.T.H.-Curtis turbine the entire casing is divided horizontally as shown in No. 17, so that the shaft and wheels may

No. 16.—Longitudinal Section of 11-Stage Curtis High Pressure Combined Impulse Turbine.

be lifted out for examination or cleaning without disturbing the alignment of the bearings.

No jointing material is necessary between the two halves of the casing, the joints being machined and scraped to an accurate surface.

No. 17.—Lower Half of Casing of 11-Stage Curtis High Pressure Combined Impulse Turbine with Wheels in Position.
6000 Kw., 1500 r.p.m.

Steam Sealing Glands.

The steam sealing arrangements at the ends of the casing usually consist of carbon rings pressed against the shaft by garter springs

(No. 18) and sealed with steam, the space inside the outer ring being maintained at a pressure slightly above that of the atmosphere. Pipes and valves are provided for admitting or exhausting the steam as may be required.

The rings press against a special sleeve forced on to the shaft and, due to the arrangement of the various stages already described, are only required to withstand approximately atmospheric pressure at the high pressure end of the turbine, and a pressure corresponding to the

No. 18.—Steam Sealing Glands of Curtis Turbine, showing Carbon Rings and Garter Springs.

No. 19.—Half of Labyrinth Packing for Large Curtis Turbines.

vacuum in the condenser at the exhaust end of the turbine. This is a feature of the greatest importance, and has a far-reaching effect on the reliability for which the Curtis turbine is conspicuous.

In some high speed machines of large capacity a metallic labyrinth packing, such as is shown in No. 19, is fitted in place of carbon packing.

Diaphragms.

The diaphragms separating the stages are constructed of cast iron. In the Curtis compound impulse turbine, on account of the large

No. 20.—Lower Half of High Pressure Split Diaphragm, showing Gland Packing and Nozzles.

No. 21.—Low Pressure Diaphragm, showing Nozzles.

amount of space available, the diaphragms can be shaped in such a manner as to give great mechanical strength, despite the fact that by comparison this machine is the shortest one on the market. In the

Curtis combined impulse turbine the diaphragms separating the single-row stages are equally strong for the pressure they are called upon to withstand, but the difference in pressure between the stages being smaller, these diaphragms are somewhat lighter. The shape of the diaphragms is well shown in Nos. 20 and 21.

The sealing of the openings in the diaphragms, through which the shaft passes from stage to stage, is effected by a patented construction consisting of an internally grooved ring split into three parts and held together by two garter springs.

The ring fits between two side-plates which prevent it moving inwards on to the shaft, but at the same time allow it free movement outwards. This effectually guards against any possible risk of damage due to inaccurate assembly and enables fine shaft clearances to be used without danger.

Nozzles.

The nozzles consist of segmental castings (No. 22) having alloy steel division plates cast into position and set at the proper angle for directing the steam against the rotating buckets.

No. 22.—High Pressure Nozzle Plate.

As the steam expands from stage to stage its volume becomes considerably greater and a larger area of steam passage is accordingly provided. This is accomplished by increasing the length of the buckets and by increasing the number and area of nozzles between successive stages.

The nozzles in the first stage extend around only a small portion of the periphery, and the intermediate or stationary buckets need extend over an area only slightly longer than that of the nozzles.

In the second and succeeding stages, however, the nozzles and stationary buckets are extended over longer and wider areas, thus accommodating greater volumes of steam. Finally, when the last stage is reached the nozzles and stationary buckets extend all the way round the periphery (No. 21).

Shaft.

The shaft is constructed of high grade steel, and designed so that the critical speed is always well above the running speed.

Wheels.

A turbine wheel running at a high speed is exposed to centrifugal forces which slightly increase the diameter of the wheel while running, with the consequent tendency for the hub of the wheel to become loose on the shaft. It is therefore necessary that the wheels should be pressed on to the shaft with sufficient force to ensure that this extension of the hub while running is less than the press fit given.

The bucket wheels are constructed of the finest steel, accurately turned and balanced, and have wide substantial wheel bosses. Each wheel, as previously explained, carries one or two rows of moving buckets, which are dove-tailed into specially-shaped grooves turned in the periphery. No. 23 shows the bucket wheels and diaphragms assembled on the shaft for a 5-stage compound impulse turbine and

No. 23.—Bucket Wheels and Diaphragms of 5-Stage Curtis High Pressure Compound Impulse Turbine.
2500 Kw., 1500 r.p.m.

in No. 24 is shown the complete rotor for an 11-stage combined impulse turbine of 6000 Kw. capacity. The disc-shaped wheels of the Curtis turbine can be much more readily constructed to resist the effect of centrifugal force than can a drum or barrel construction; while the very large section of boss possible in the Curtis design entirely obviates the danger of a wheel becoming loose on the shaft when fully stressed.

A loose wheel will set up injurious vibrations very adverse to the life of the machine, so that the importance of the B.T.H. design of bucket wheel in this respect cannot be over-estimated.

In pressing a steel wheel on a steel shaft with the requisite force it has been found that the shaft is often torn and damaged owing to the excessive friction between steel and steel. In order to obtain the

proper fit and avoid this trouble, all B.T.H.-Curtis turbine wheels are provided with two bronze bushes pressed into the hub of the wheel. These bushes enable the necessary force to be used without the danger of tearing or damaging the shaft.

No. 24.—Rotor of 11-Stage Curtis High-Pressure Combined Impulse Turbine.

6000 Kw., 1500 r.p.m.

The wheels are pressed on and taken off the shaft by means of a special hydraulic jack which screws on to the turbine shaft. The hydraulic cylinder which surrounds the shaft is reversible in its action, so that the wheels can be put on or taken off by the same jack.

No. 25.—Bucket Wheels and Diaphragms in Position in Casing of 5-Stage Curtis Compound Impulse Turbine.

No. 26.—Bucket Wheels and Lower Half of Split Diaphragms in Position in Casing of 5-Stage Curtis Combined Impulse Turbine.

Buckets.

The moving buckets are of special steel or bronze section, highly finished. Each bucket is milled at one end to a dove-tail, as shown in No. 27, so as to correspond with the grooves in the periphery of the wheel. The buckets are inserted through an enlarged opening in the groove, and are assembled with distance pieces placed between them in the manner shown in No. 28. When all the buckets are in position the opening is filled with a caulking block and closed up. A shroud ring is also riveted to their outer ends to keep the flow of steam inside the bucket area.

No. 27.—Buckets for Curtis Turbines.

The stationary buckets ("intermediates") which are used for reversing the direction of motion of the steam are, where used, arranged between successive rows of moving buckets. They are constructed in a similar manner to the moving buckets and fixed in segments, as shown in No. 29, which are securely bolted to the turbine casing or diaphragm.

Adjustment of Clearances.

The clearances between stationary and moving buckets can, if desired, be measured by removing sight-hole plugs in the casing. A simple screw device is provided in connection with the thrust

block whereby the clearances can be adjusted to the utmost nicety while the turbine is running.

Thrust Block.

As previously explained, the amount of thrust in a Curtis turbine is practically negligible, but some anchoring device is required to

No. 28.—Method of Inserting Buckets in Groove of Wheel.

keep the successive rows of moving and stationary buckets in their correct positions relatively to each other. A thrust block of the usual form is found to be the best device for this purpose, although the actual thrust on a block is inappreciable. No. 30 shows the lower half of a typical thrust block fitted in place on the end of the bearing liner. The thrust block consists of a split cast-iron sleeve lined with white metal to fit grooves turned directly in the shaft. These grooves are clearly shown on the shaft in No. 24.

No. 29.—Section of Stationary Buckets (Intermediates).

Bearings and Lubrication.

In the B.T.H. turbo-generator, the turbine and its generator have each two bearings, their shafts being connected by means of a double-claw flexible coupling. This is made up in four parts and so arranged that by taking out a few bolts either the turbine or generator rotor may be removed without disturbing the other.

No. 31 shows a complete coupling bolted up, and No. 32 is a view of a large coupling capable of transmitting 10000 H.P., showing

No. 30.—Position of Thrust Block on Liner.
High Pressure End Bearing.

No. 31.—Flexible Coupling.

No. 32.—Flexible Coupling to Transmit 10000 H.P.

the four parts disconnected. The coupling is contained in one pedestal which carries the two middle bearings, the claws being lubricated with oil from the latter.

The exciter armature shaft is also connected to the alternator shaft by a flexible steel disc coupling, so that all three shafts of the unit are independent of each other for alignment, and satisfactory running of the machine is assured even if the bearings wear unevenly or if the true alignment be disturbed through settling of the foundation.

The bearing at the high pressure end of the turbine also has attached to it the thrust block as shown in No. 30. Both turbine and generator bearings consist of cast-iron shells lined with babbit metal, and are spherically seated and self-aligning (No. 33). They are lubricated by oil under pressure from a valveless rotary oil pump running at a low speed, which, together with the governor described below, is driven by a worm and worm wheel from an extension of the main turbine shaft beyond the thrust block.

No. 33.—Spherical-Seated Bearing.

No. 34.—Oil Pump. Cover Removed, showing Wheels.

The pump consists of two small toothed wheels gearing with one another and fitting exactly into an iron casing (No. 34). When running, the oil is urged round between the wheels and case, the oil filling the spaces between adjacent teeth. As there are no valves or springs, etc., the pump is practically immune from breakdown.

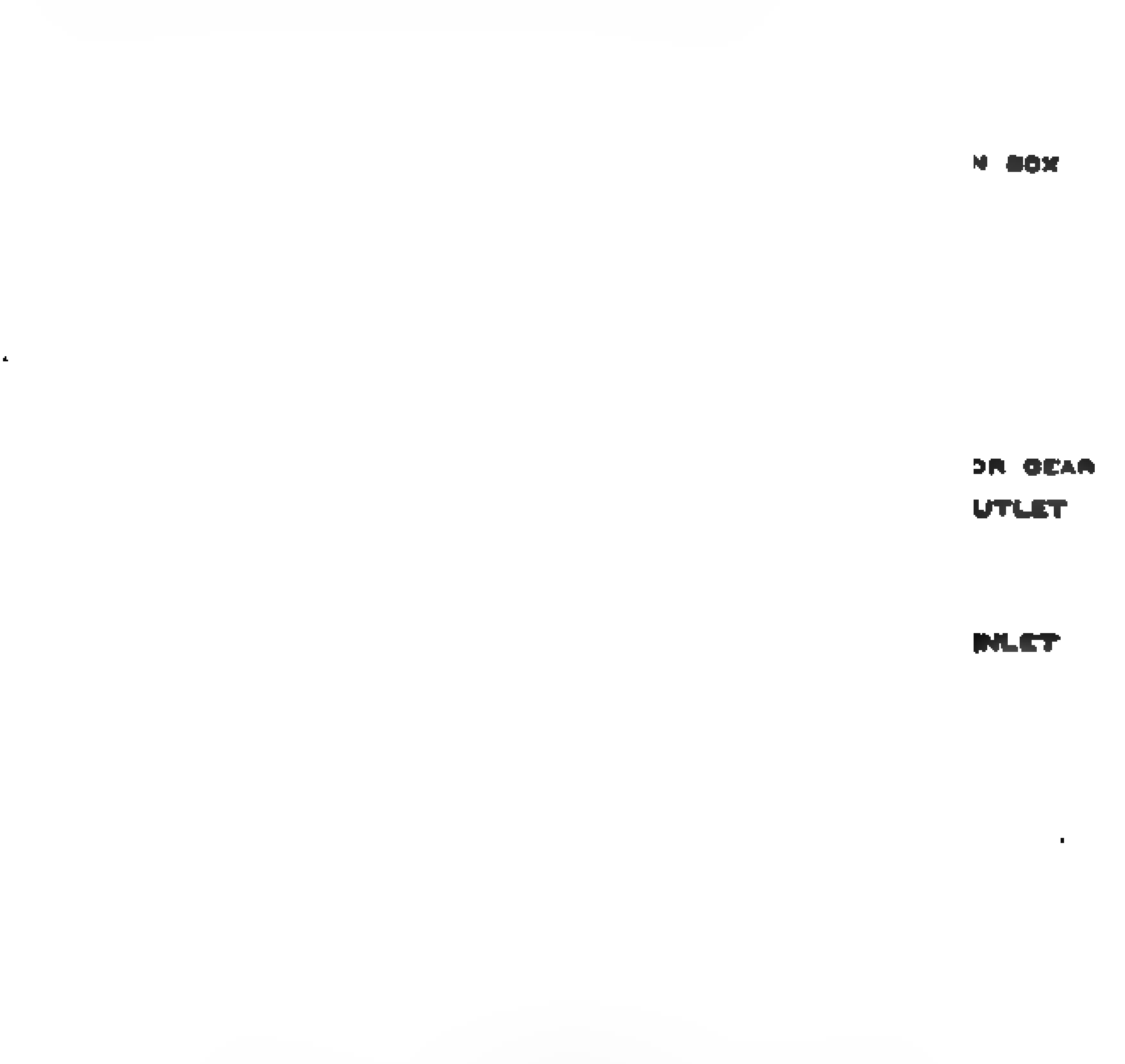
The oil, after leaving the bearings, returns to a large reservoir and thence to an oil settling tank wherein any water carried by the oil can be separated out. In the suction pipe, between the reservoir, and pump, a strainer is placed to arrest any foreign matter; this strainer can be cleaned or renewed without stopping the supply of oil to the pump.

The oil is also cooled on its way to the bearings by passing through an oil cooler (No. 35) containing tubes through which cold water is circulated, and amongst which the stream of oil is distributed with consequent loss of heat.

As shown in No. 35, an oil return box and regulating valve is provided on each bearing, by which the quantity and temperature of the oil passing through it can be observed and regulated.

On the smaller sizes of turbines, from about 1500 Kw. downwards, the bearings are provided with oil rings which help in the distribution of the oil and enable the turbine to be started without flooding the bearings—an ample well being provided under each bearing to hold a large quantity of oil for the rings.

On larger turbines, an auxiliary hand or steam pump is provided for priming the lubricating system before starting.



No. 35.—Oil Circulating and Cooling System of Curtis Turbine.

On the very large sizes an oil accumulator is fitted for the purpose of preventing excessive variation in the oil pressure during heavy changes in load.

Bedplate.

Both turbine and generator are mounted on a combined cast-iron bedplate of heavy box section, which ensures very great rigidity and the maintenance of correct alignment.

So far as is consistent with ease of handling and transport, the bedplate is always constructed in one piece.

No. 36.—Turbine Governor.

Part Casing Removed, showing Weights and Springs.

Governor.

The governor (No. 36) is of the centrifugal type, and is carried on a vertical shaft driven from the same source as the oil pump just referred to, although in the largest sizes the oil pump is driven by separate gear.

To ensure the most sensitive and reliable action, the governor is designed so that the work performed by it is as small as possible, and for this reason it is not arranged to actuate the main steam control valves direct.

The governor, instead, controls a small balanced pilot valve which admits oil under pressure of about 40 lbs. per square inch to one side or the other of a rotary piston in a servo-motor, which is the real agent in controlling the main steam control valves.

The governor is adjusted so as to control the steady speed of the turbine within 2½ per cent. from no load to full load, and is provided with a hand-wheel for adjusting the compression of a balancing spring, by means of which the speed of the turbine may be varied 5 per cent. either above or below normal while the machine is running.

Valve Chest and Controlling Gear.

Steam is admitted to the turbine through a number of fixed nozzles attached to the main valve chest. These nozzles are divided into groups, each group communicating through a separate passage with valves, situated in the steam chest. By opening or closing these controlling valves and thus keeping the length of the steam belt always proportional to the load, the amount of steam going to the turbine is regulated.

The number of valves varies with the size of the machine, the smallest machines being fitted generally with about four valves, while the largest machines may have up to ten. The valves are lifted off their seats by means of a cam shaft mechanism, and they are closed by an opposing spring force.

The springs are arranged either within the steam space (when the temperature of the steam permits) or placed outside, where, although they take up more room, they are not exposed to a high temperature.

The valve chest is made of cast steel for all pressures from 150 lbs. upwards. For lower pressures than this, cast iron has been found to be satisfactory.

Action of Governing Gear.

The motion of the governor is transmitted through the floating (differential) lever, to the piston valve which admits the oil to one side or the other of the rotary piston, and at the same time opens the exhaust port on the opposite side (No. 38). When the valve is in its central position, covering up both ports, the servo-motor is locked.

The motion of the rotary piston rotates the cam shaft and also the small toothed pinion gearing into a rack. The upper end of the rack is attached to the floating lever as shown.

No. 37.—Valve Chest (at top) and Governor Gear Assembled on Curtis High Pressure Turbine.

Any motion of the governor end of the floating lever moves the piston valve from its central position, with the result that oil under pressure enters one port and exhaust oil returns from the other.

The rotating piston moves through just the angle necessary to restore the piston valve, through the medium of the rack, pinion, and floating lever, to its central position.

No. 38.—Governing System of High Pressure Curtis Turbine.

The motion of the servo-motor is always some fixed proportion of the governor motion, hence an exactly proportionate motion to that of the main governor is produced by the servo-motor, with the full pressure of the oil behind it.

No. 39.—Emergency Governor Ring.

The cam shaft has upon it a series of cams which successively operate steam poppet valves as the rotary piston describes its stroke, until all valves are open.

Each poppet valve admits steam to a section of the first stage

No. 40.—Emergency Governor Ring in Place on Shaft.

nozzles of the turbine. By this method of governing (opening nozzles in succession) there is no loss due to throttling of steam except in the poppet valve, which at any instant may be under the operation of the governor, and full steam pressure is always applied at all loads.

Emergency Governor.

A speed limiting device is provided with all B.T.H.-Curtis turbines, whereby, in the event of an accident occurring to the main governing mechanism causing the speed to rise above the normal, the steam is automatically cut off before a dangerous speed is reached.

The emergency governor consists of an unbalanced steel ring (No. 39), held by means of a helical spring concentrically with the turbine shaft (No. 40) so long as the speed of the shaft remains normal. Should the speed exceed a predetermined value, however, the centrifugal action will overcome the spring and cause the ring to fly out eccentrically. In this position the ring comes into contact with a trigger which trips the valve mechanism, and thus allows the emergency valve to close instantaneously.

CURTIS LOW PRESSURE STEAM TURBINES.

(Mixed Pressure Type.)

Expansion of Steam.

It is a fact not generally realised that the energy liberated by the expansion of a given weight of dry steam depends upon the number of times that the original volume is increased or upon the ratio of the pressure drop, rather than the actual pressure drop. For example, a pressure drop from 30 lbs. down to 15 lbs. absolute liberates very nearly the same amount of energy as a pressure drop from 200 lbs. to 100 lbs. The pressure ratio is the same in both cases, but the pressure difference is very much greater at the higher pressures.

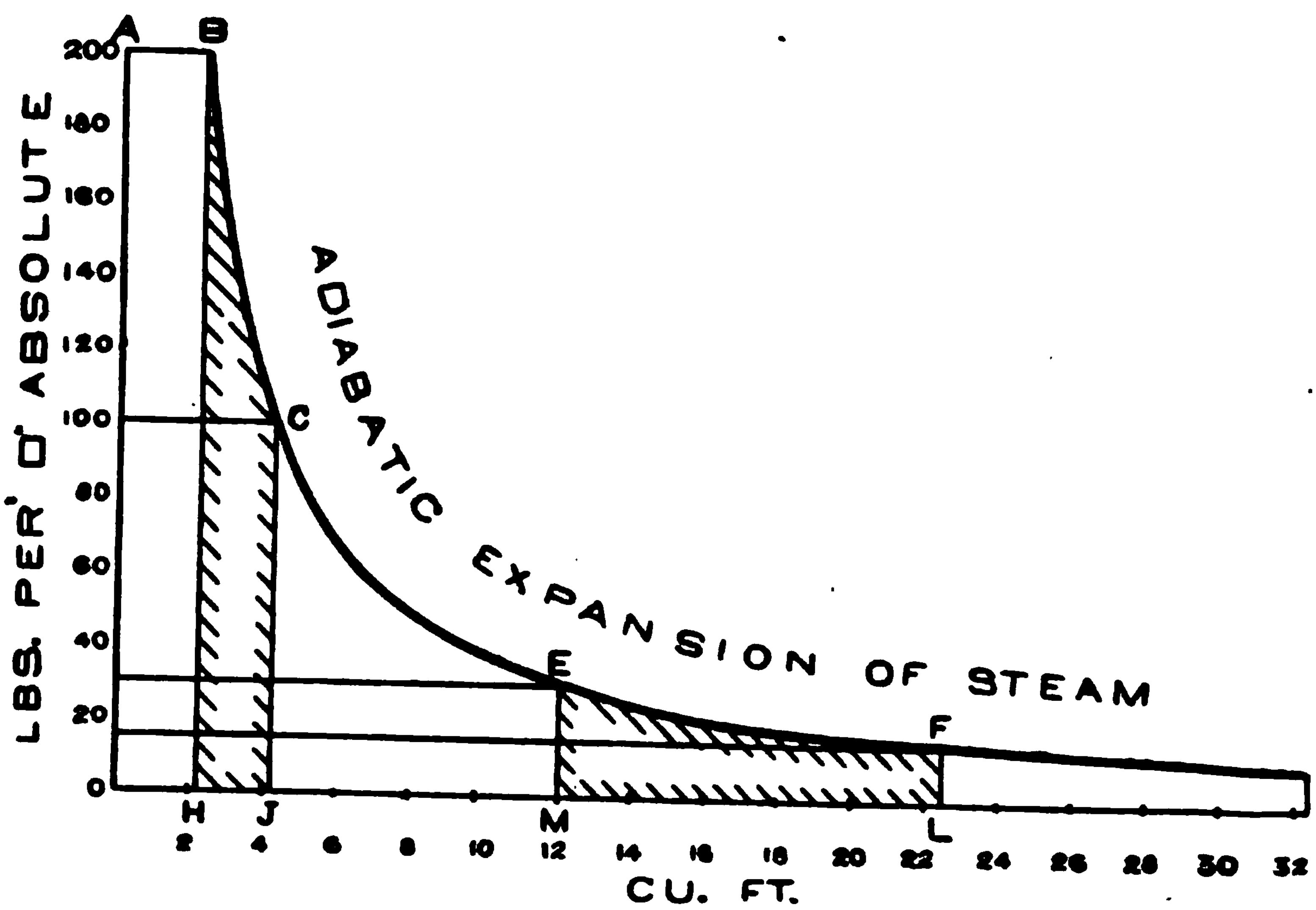
In No. 41 the curve represents diagrammatically the adiabatic expansion* of 1 lb. of dry steam from 200 lbs. down to 10 lbs. absolute, and is limited only by lack of space from continuing the process further. The steam pressure is measured vertically, and the volume horizontally from the point O, so that the area of the diagram (which is the product of its length and average height) is proportional to the energy developed, this being the product of pressure by change of volume.

The volume occupied by 1 lb. of dry steam at 200 lbs. absolute pressure is 2·288 cub. ft., which is represented by the position of the letter H in the diagram. If now the steam is expanded down to 100 lbs. (C) the volume will be increased to the point J, *i.e.*, by 84 per

* Expansion without loss or gain of heat to or from surrounding bodies.

cent., and the amount of energy liberated, assuming the back pressure to be zero, will be denoted by the shaded area BCJH.

In the same way the area EFLM shows the work done by the expansion of the steam from 30 to 15 lbs. This area is less than the area BCJH, but is not nearly as much reduced as the pressure drop has been reduced. The fall of pressure in the first case is 100 lbs., whereas in the latter case it is only 51 lbs. The process of halving the pressures, etc., can still be continued, and it can be shown that the adiabatic expansion of 1 lb. of steam from 2 lbs. to about 0·64 lb. absolute, *i.e.*, from a vacuum of approximately 26 in. to one of 28·7 in.



No. 41.—Diagram showing Adiabatic Expansion of Steam.

of mercury (with barometer at 30 in. Hg.), liberates just as much energy as the expansion of the same weight of steam from 200 lbs. to 100 lbs. absolute.

As a matter of fact, in the steam turbine expansion takes place in accordance with a curve which lies slightly higher than the adiabatic curve, so that there is actually more energy liberated in the lower expansions than is shown in No. 41. The point at issue, however, will be clear, namely, that a very large amount of energy is available in steam at low pressures.

Unsuitability of the Reciprocating Engine for Low Pressures.

The volumes of steam corresponding to the low pressures just mentioned are very great, and for this reason reciprocating engines cannot be adapted to accommodate them. The mechanical diffi-

culties presented by the undue size of the low pressure cylinders and valve gear necessary, preclude any advantage being realised with a vacuum higher than 26 in.

Suitability of the Turbine for Low Steam Pressures.

Turbines, however, due to their inherent design and the absence of slide valves and ports, together with the fact that in them the steam is constantly in motion, can be designed to accommodate very large volumes of steam at the lowest pressures, without entailing any great increase in cost.

For this reason turbines can be used to great advantage for running on the exhaust steam from reciprocating engines or other steam using devices, and extracting thereby a very considerable amount of the energy in the steam by expanding it down to the lowest practicable limit.

Available Energy in Steam.

The appended tables show the available amount of energy in steam in British Thermal Units (B.Th.U.) under various pressure ranges. One B.Th.U. is equal to 778 foot lbs. of work.

BRITISH THERMAL UNITS AVAILABLE WHEN STEAM IS EXPANDED FROM A HIGHER TO A LOWER PRESSURE.

AVAILABLE ENERGY IN B.TH.U. PER LB. OF SUPERHEATED STEAM.

100° F. Superheated Steam at Lbs. per Sq. In. Absolute Pressure.	Available Energy in B.Th.U. when Expanded to :—				150° F. Superheated Steam at Lbs. per Sq. In. Absolute Pressure.	Available Energy in B.Th.U. when Expanded to :—			
	14.7 Lbs. per Sq. In. Absolute Pressure.	27 In. Vacuum.	28 In. Vacuum.	28½ In. Vacuum.		14.7 Lbs. per Sq. In. Absolute Pressure.	27 In. Vacuum.	28 In. Vacuum.	28½ In. Vacuum.
250	222	352	373	386	250	229	364	384	397
200	203	337	358	371	200	212	348	369	382
175	193	328	349	362	175	202	339	360	374
150	182	318	339	352	150	190	328	349	363
125	168	305	327	340	125	175	315	337	351
100	151	290	312	326	100	158	299	321	335
90	143	282	304	318	90	149	291	313	327
80	134	274	296	310	80	140	283	305	319
70	123	265	287	301	70	129	273	295	309
60	111	254	276	290	60	117	262	284	298
50	97	241	263	277	50	103	250	272	286
40	80	226	248	262	40	85	234	256	270
30	58	206	229	244	30	62	213	236	251

Available Energy in B.Th.U. per Lb. of Dry Steam.

Dry Steam at Lbs. per Sq. In. Absolute Pressure.	Available Energy in B.Th.U. when Expanded to:— 14.7 Lbs. per Sq. In. Absolute Pressure.	27 In. Vacuum.	28 In. Vacuum.	28½ In. Vacuum.	Dry Steam at Lbs. per Sq. In. Absolute Pressure.	Available Energy in B.Th.U. when Expanded to:— 14.7 Lbs. per Sq. In. Absolute Pressure.	27 In. Vacuum.	28 In. Vacuum.	28½ In. Vacuum.
250	204	330	349	362	95	136	269	290	304
245	203	329	348	361	90	132	266	287	300
240	202	328	347	360	85	128	262	283	296
235	201	327	345	358	80	124	258	279	292
230	199	325	344	357	75	119	254	275	288
225	198	324	343	356	70	114	248	270	283
220	196	322	341	354	65	108	243	265	278
215	194	321	340	353	60	103	238	259	272
210	192	320	339	352	55	96	233	254	267
205	191	318	337	350	50	89	227	248	262
200	189	316	336	349	45	81	220	241	255
195	188	315	334	347	40	74	213	233	247
190	186	313	332	345	35	63	204	225	239
185	184	311	331	344	30	51	194	215	229
180	182	310	329	342	25	39	182	203	217
175	180	308	328	341	20	22	168	190	204
170	178	306	326	339	19	...	164	186	200
165	176	304	324	337	18	...	160	182	196
160	174	302	322	335	17	...	157	179	193
155	172	300	320	333	16	...	153	175	189
150	169	298	318	331	15	...	149	171	185
145	166	296	316	329	14	...	144	167	181
140	164	294	314	327	13	...	139	162	177
135	162	292	312	325	12	...	134	157	172
130	159	290	310	323	11	...	128	151	166
125	156	287	308	321	10	...	122	145	160
120	153	285	305	318	9	...	115	138	153
115	150	282	302	315	8	...	108	131	146
110	146	279	300	313	7	...	100	123	138
105	143	276	297	310	6	...	89	113	128
100	140	273	293	307	5	...	79	102	117

From the table it will be seen that when dry saturated steam is expanded from a boiler pressure of 150 lbs. gauge (165 lbs. absolute) to atmospheric pressure (14·7 lbs. absolute), the available energy or work amounts to 176 B.Th.U. per lb. weight of steam, and when further expanded from atmospheric pressure to a vacuum of 28 in (about 1 lb. absolute) the extra available energy in B.Th.U. is about 148 B.Th.U. per lb.

It will be seen from the above that under these very usual conditions there is theoretically nearly as much work available in the low pressure range as in the high.

In actual practice, allowing for condensation in the exhaust pipe, the output obtainable from the exhaust steam is usually about 75 per cent. of the output of the existing engine—when this is of a fairly economical type—this amount naturally increasing with the wastefulness of the engine.

The above, of course, applies to a case in which a low pressure turbine is operated from the exhaust steam of an existing non-condensing engine. There is also, however, a very material economy to be obtained even when the existing plant is operated condensing. The reason for this is that in most condensing engines the saving over non-condensing conditions does not exceed 25 per cent. to 30 per cent. under the most favourable conditions of load, and on overloads the saving by condensing is much smaller. It has been found in many cases where exhaust turbines have been applied to existing engines that the total output has been increased some 30 per cent. to 50 per cent. without any increase in the fuel consumption.

Under overload conditions, where the engine efficiency falls off and where its gain by vacuum is greatly diminished, the rate of improvement is even better than this.

Application of Turbines to the Utilisation of Low Pressure Steam.

Turbines can be advantageously applied in most cases where reciprocating engines are already in use, and their application will effect a large economy and increase of existing output. This applies whether the existing engines are at present operated condensing or non-condensing, and also to engines which are operated intermittently, as in the case of colliery winding engines, rolling mill engines, etc.

Exhaust *versus* Mixed Pressure Turbines.

The purely low pressure or exhaust turbine is not advocated by the B.T.H. Co., as the mixed pressure machine, with its many advantages, can be obtained with only a small increase in cost. Practically the only situation where a pure exhaust turbine is justified is where it is to be connected to one engine, with which it operates as one unit controlled only by the engine governor.

The great advantage of the mixed pressure over the exhaust pressure turbine lies in the fact that not only does it make the same use of the exhaust steam as in the pure exhaust type, but also, in addition, will carry all or any portion of its rated load on high pressure steam direct from the boiler. Special high pressure nozzles are provided which automatically come into action in case the supply of low pressure steam is for any reason insufficient for the power required from the turbine.

BRITISH WESTINGHOUSE RATEAU-IMPULSE TYPE.

The following notes, descriptions, and illustrations of Rateau type of impulse turbines and turbo-generators are reprinted by special permission of the British Westinghouse Electric and Manufacturing Company Limited, Trafford Park, Manchester, who hold the British licence for the manufacture of the turbines mentioned.

Descriptive.—An introductory description of the principles underlying the operation of steam turbines will be useful, especially to those who have only a very superficial knowledge of the subject, particularly to enable them to draw some comparisons between the terms "Reaction" and "Impulse" as applied to turbines.

In the first place, any body, whether it be in a solid, liquid, or gaseous form, possessing weight and velocity, can be brought to rest only by giving up energy, as conversely, energy must be expended in giving it velocity. The amount of energy in each case is the same.

Steam, in a perfectly dry state, is a gas, but directly it does work in any form a portion turns to water, so producing a mixture. It is capable of giving to itself velocity by dropping in pressure and expanding in volume; in other words, it changes some of its intrinsic heat or potential energy into velocity or kinetic energy. In this way it does work and becomes wet.

In an "Impulse" turbine the steam falls in pressure through nozzles, thereby gaining considerable velocity. The jet of steam at this high velocity is directed against the moving blades, on which the work is performed, and the steam is thereby brought to rest.

In the "Reaction" turbine the principle is quite different, though the steam attains its velocity in exactly the same manner. To return to the nozzle again, let us consider that the nozzle is not fixed in any way, and that any force exerted upon it can push it in any direction without resistance. If the steam were expanding freely it would radiate in all directions from the centre of pressure, but since it flows in one direction only, some force must have been at work to confine it to this one direction, and this force will move the freely suspended nozzle in the opposite direction to that in which the steam flows.

Referring to Nos. 42 and 43 which illustrate this:

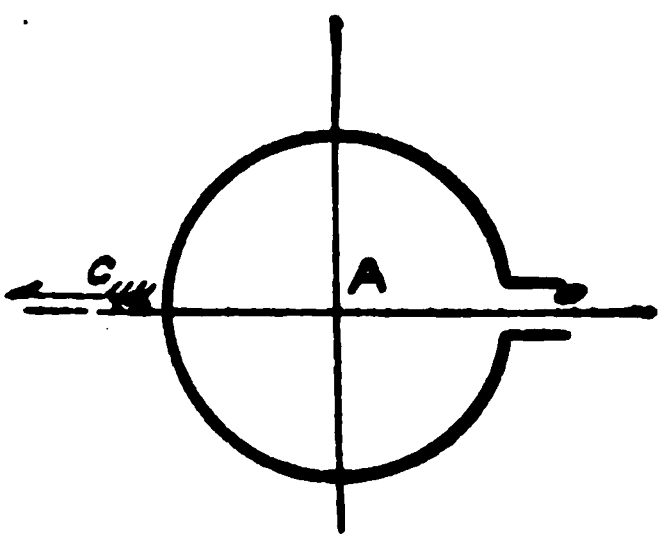
Suppose A to represent a spherical vessel containing steam at say 200 lbs. per square inch pressure, and having an opening or nozzle open to the air. The pressure against every square inch of the inside of the vessel will be 200 lbs., with the exception of the area cut out by nozzle, which, at its throat *a* will have a pressure of only 116 lbs. If the area of *a* is 1 sq. in. there will be a pressure of 84 lbs.,*

* It is a well-known law that the terminal pressure has no effect upon the pressure at the throat of a nozzle below 0·58 of the initial or higher pressure, therefore the pressure in the instance given will be $0{\cdot}58 \times 200 = 116$ lbs. per sq. in.

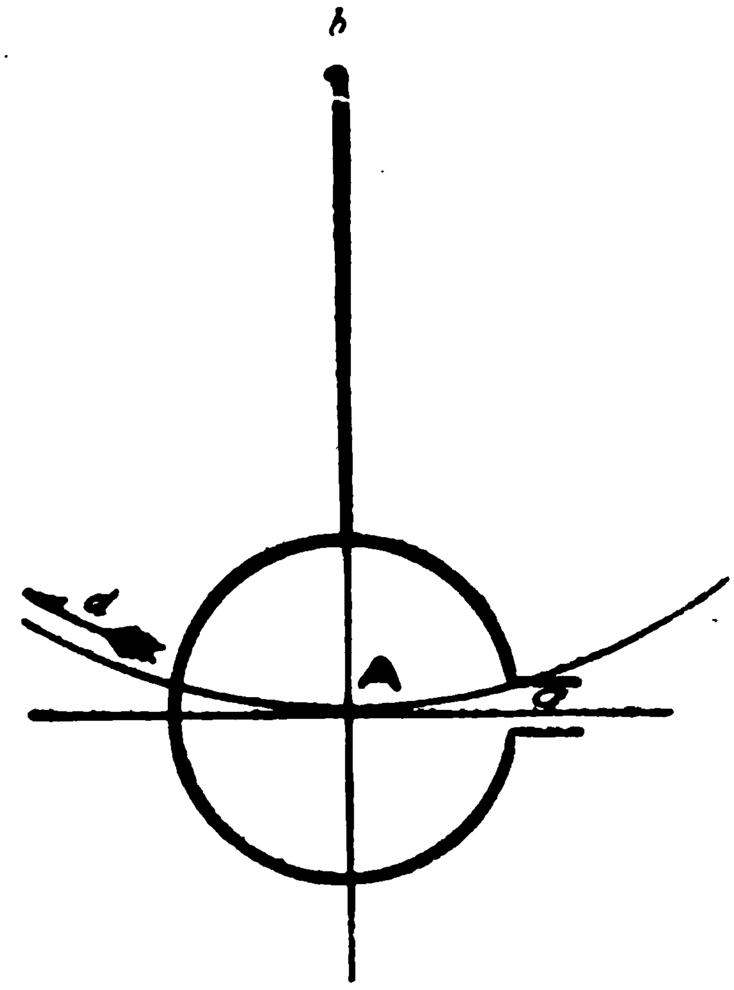
tending to thrust the vessel in the direction indicated by the arrow *c*. If the vessel were tied to a point *b* in such a manner that it could revolve round this point, then the vessel would revolve in the direction indicated by the arrow *d* and would be capable of doing work.

This principle was employed in the earliest turbines of which we know, and dates back to the time of Hero, 2000 years ago.

The modern reaction turbine consists of a number of nozzles round the whole circumference of the turbine, acting in a similar manner to that illustrated above. As the drop in pressure, however, is generally too great to be utilised in one set of nozzles, a number



No. 42.



No. 43.

are usually placed in series, and the fall in pressure through the turbines is carried out gradually.

It is evident that this reaction principle is entirely different from the impulse principle; the former drives the blades or nozzles in the opposite direction to that in which the steam is flowing, but in the latter the steam flows in the same direction.

A very common instance, involving the principles of reaction and impulse, is that of a falling ball. In falling, the ball loses potential energy and gains kinetic or moving energy, which reaches a maximum at the instant the ball strikes the floor. It strikes it with an impulse, and from the instant it does so until it ceases its downward movement the kinetic energy is gradually absorbed in deforming small portions of the ball and the floor. When it is entirely absorbed, the ball being elastic, begins to regain its original shape, which it can do only by forcing the ball upwards as the deformed part is pressed on the floor. This is reaction, and the height to which the ball will be pro-

jected will depend upon the height through which the ball has fallen, and the elasticity of the ball and the floor.

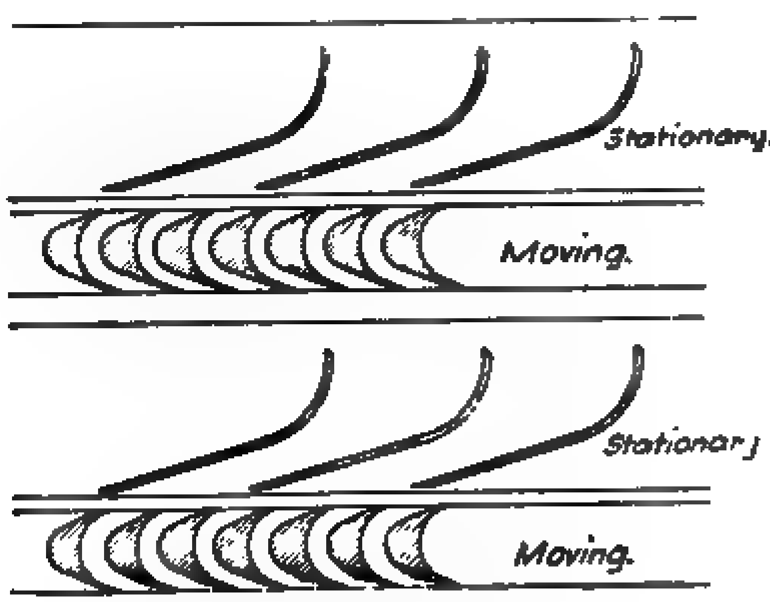
It will be evident that no turbine can be efficient which fails to use all the velocity in the steam, for if velocity still exists it is capable of doing work; in other words, it is essential that the steam after striking the blades should have no velocity in relation to the casing. For this to exist in the case of the impulse turbine the steam must have a velocity twice that of the blades, since after striking, the steam must rebound from the blades with the same velocity as that at which the blades are travelling. Now in the case of the reaction turbine the steam, to have no velocity in relation to the casing, must drive the blades backwards with the same velocity as that at which the steam is travelling. The number of stages of a reaction turbine must therefore be twice as many as an impulse machine for the same blade speed.

One of the chief factors contributing to distortion of the spindle or the cylinder lies in the fact that in the reaction turbine steam is admitted into the casing at practically full boiler pressure and at very high temperature. Of late years, manufacturers of this machine have introduced a velocity wheel at the high pressure end. This enables them to construct the machine for "nozzle control," and at the same time to reduce the steam pressure and temperature to which the cylinder proper is subjected. Even this method, however, does not overcome all the difficulties, for in obviating to a certain extent those mentioned above, it leaves the machine open to another objection, which is, that the relative velocity of the steam and the blades is so vastly different in the velocity wheel and the drum, *i.e.*, in the impulse principle and in the reaction principle, that it is necessary for the velocity element to be considerably larger in diameter than the reaction portion. The result is that the cylinder cannot be made of a uniform diameter throughout, and in addition to being difficult to cast, is such that it cannot be relied upon to expand or contract evenly throughout its whole length. From this it will be seen that the solution of the problem is by no means met by this construction.

Turning now to the type of impulse turbine made by the British Westinghouse Company, they, while recognising the advantages accruing from the use of a velocity wheel at the high pressure end, do not consider it advisable, except in very special circumstances, to fix this type of wheel at the low pressure end. As is now well known, in order to get a maximum economy from this it is necessary to employ very high steam speeds. High, however, as these steam speeds are, there can be no fear of blade erosion as long as the steam is dry. In the operation of the machine the steam must of necessity become wet at the exhaust, due to the work done by the steam. The British Westinghouse Company, to deal with this, employ, at the low pressure end, stages requiring a lower steam speed than that necessary for the velocity compound wheel. This condition was found to be met by the Rateau impulse wheel, the blading of

which is illustrated in Nos. 44 and 45. With this it is only necessary for the steam speed to be about one-half that required for the velocity compounded element. At the same time, the clearances may be made amply large, since no expansion takes place in the moving blades.

With regard to the pure Rateau type of turbine, the British



No. 44

No. 45

Westinghouse Company, after careful investigation and experiment, decided that, although there are a few special cases where it is desirable, no advantage is to be derived from its general adoption. A study f No. 46 will make the following remarks quite plain to those who have only a superficial knowledge of steam turbine design. No. 46 shows in the lower half a longitudinal section through the impulse turbine as now manufactured by the British Westinghouse Company and on the chart on the upper half will be found a diagram

showing the change of pressure and temperature throughout the machine, and the velocity of the steam in the various stages. Attention is specially directed to the pressure and temperature of the steam after the first stages of each of these types.

The machines illustrated in this diagram have been calculated for an initial steam pressure of 175 lbs. per square inch absolute, superheated to a total temperature of 525° F., and for an exhaust pressure of 1 lb. absolute.

It will be seen that the velocity wheel at the high pressure end is a very desirable feature, since the steam pressure is reduced to 35 lbs. per square inch absolute, and the temperature to 265° F. before being admitted to the cylinder proper. With the pure Rateau type of machine, the pressure inside the casing would be as high as 85 lbs. per square inch absolute, and the temperature 415° F.

To obtain the same efficiency as with the Curtis Rateau turbine shown in diagram No. 46, it would be necessary to fit three Rateau wheels in place of the velocity wheel. For the same efficiency, the machine with the velocity wheel at the high pressure end is shorter between the bearings than the pure Rateau machine. The shaft will consequently be much stiffer and the critical speed higher.

While the drum turbine is cheaper to construct, the disc type lends itself to more accurate calculation of the stresses. Each wheel can be balanced separately on the shaft, and thus the whole spindle is balanced throughout. The drum construction may be balanced statically, but be considerably out of running balance.

These are the reasons why the British Westinghouse Company adopted as their standard the pure impulse turbine, having a velocity wheel at the high pressure end and the Rateau impulse wheels for the succeeding stages.

High Pressure Turbines.

The expansion of the steam from boiler to condenser pressure is carried out in successive stages, the number of which depends upon the pressure range and relative steam and blade velocity.

Referring to No. 48, the steam first expands through nozzles in one or more of the nozzle blocks where potential energy in the steam in the form of pressure and superheat is converted into velocity or kinetic energy. This first transformation gives to the steam sufficient velocity to enable it to do work efficiently on two rows of moving blades. After striking the first row it has sufficient reserve velocity to do a large amount of work in the second row after passing through the deflecting or guide blades formed in a row between them. The first wheel is therefore compounded for velocity, and is known as the velocity wheel: it is shown at *m*. A higher efficiency is obtained by adopting this method of dealing with the steam in its initial stage than is possible with any other construction. The reverse is the case at the final stages. Another point of perhaps even greater importance

No. 47.—General View of a High Pressure Westinghouse-Rateau Impulse Turbine.

"The Marine Steam Turbine." [To face page 549.

is that most turbine troubles are attributable to unequal expansion and contraction of the cylinder and spindle under changes of temperature by fluctuating steam pressure, superheat, vacuum, and load, thus causing distortion. It is therefore advisable to reduce the temperature range as much as possible. With the Westinghouse construction the temperature of the steam inside the cylinder is reduced to little more than half its total temperature at the stop valve. The heat value of the steam is converted into work in expanding through the nozzles. The full temperature of the steam is therefore confined to the nozzle blocks. All the large turbines which the British Westinghouse Company have designed and placed upon the market employ this principle.

After the steam leaves the velocity wheel its expansion is continued in single stages until the exhaust pressure is reached. Each of these stages consist of a diaphragm fixed in the cylinder and a moving wheel on the shaft. The velocity of the steam by expanding through the diaphragm nozzles is completely used up on the moving wheel. Before passing into the cylinder the steam goes through various valves which will be described under the heading "Steam Chest."

No. 48 is a longitudinal section through such a turbine. The discs carrying the moving blades are shown at *aa*, and the diaphragm with fixed blades or nozzles at *bb*. The nozzle blocks through which the steam first expands are shown at *c*. These are generally three in number; one will take up to half-load, two up to full load, and the third is for overload or running non-condensing. Two of these nozzle blocks are generally arranged in such a manner that they may be cut out if desired, when running on small loads for a long time. Special glands *dd* are provided at each end of the cylinder.

The bearings *ee* are separated from the glands by small isolating chambers which separate the cylinder from the bearing housings and casings in which the parts requiring lubrication are situated, and which are best kept cool. The covers of the bearing housings are arranged so that they can be quickly and easily removed without disturbing any other portion of the machine. To maintain the spindle in its correct relative position with the cylinder, a registering block *f* is provided. Next to this on the spindle is a worm driving a worm wheel which in turn drives the main governor. On the extreme end of the spindle is the safety governor *h*. At the other end of the spindle is the coupling *k*. The oil pump, driven by a vertical shaft through bevel gears from the main governor shaft, is placed in the oil tank at the end of the bedplate. This pump delivers the oil under pressure to the cooler, from whence it is forced to the various bearings. After passing through the bearings it flows back by gravity to the oil tank, so completing the cycle. A small auxiliary oil pump is also fitted for starting up.

No. 48.—Longitudinal Section through High Pressure Westinghouse-Rateau Impulse Turbine.

Cylinder.—Special attention is given to the selection and mixing of the cast iron to ensure the closeness of grain and homogeneity necessary for the best cylinder castings. Though in this type of turbine the cylinder is not subjected to more than about half the maximum temperature to that which the reaction type is subjected, every precaution is taken in design to reduce distortion by heat to a minimum. A judicious use of ribs gives the stiffness desired, but at the same time they are so proportioned that they do not constitute a weakness by having such differences of temperature between the parts as to cause them to crack. One end of the cylinder is fixed to the bedplate, and the other end is free to slide on a guide to allow of expansion. Again, the expansion of the cylinder in a direction at right angles to the axis of the turbine is equally divided on each side, special guides being provided for this purpose. To enable the top half of the cylinder to be removed without fear of damaging the spindle, guide rods are fixed to one half and slide through long guide holes in the other.

Nozzles.—The high pressure nozzles are built up of special forged steel, machined all over to ensure accuracy. This method is much superior to that in which cast nozzles are employed, as in the latter the area angles of the nozzles cannot be controlled with the degree of precision necessary to ensure efficiency and longevity.

Diaphragms.—These are made of either cast steel or cast iron depending upon circumstances. From No. 49 it will be seen that they are in halves, and provided with spigot and socket joints to prevent leakage of steam from one side of the diaphragm to the other. At B is shown a diaphragm having partial admission, and at C another with full admission. In the earlier stages of the expansion of the steam its volume is comparatively small, so that to admit steam through the diaphragm nozzles all round the circumference would curtail their height, and inefficiency would result.

It will be noticed that these diaphragms are arranged in such a manner that the respective halves in cylinder cover and lower half are not disturbed when the cover is lifted. The outer periphery fits in a groove in the cylinder, and the inner periphery is provided with special glands which almost touch the shaft. They are easily renewable, and of such a nature and design that if touching takes place between them they wear their own clearance without causing any trouble. The nozzles or guide-blades are made of a special steel, and cast into the cast-iron diaphragms by a special process, the development of which has enabled the British Westinghouse Company to make diaphragms of a very superior type. When cast-steel diaphragms are used these nozzle blocks are segmentally fixed to them in a thoroughly secure manner. The advantage of the Westinghouse system of split diaphragms cannot be overlooked in regard to

accessibility in examination. This should be contrasted with the method adopted by other makers of similar types of turbines, where

No. 49.

the diaphragms are in one piece and threaded on to the shaft alternately with the blade wheels. In the latter case it is necessary to dismantle practically the whole turbine in order to examine the diaphragm glands.

e Wheels.—These are made of the finest grade wrought ›ering in thickness from the centre to the periphery where ‹ness is again increased and formed to take the blades. This ; gives the greatest strength and stiffness consistent with ›s, though the cost of manufacture is necessarily heavy. Each inely machined all over, is provided with large balance holes ıre the pressure at each side always being the same. After ›laded complete, each wheel is carefully balanced statically to ıe nearest possible approach to an ultimately perfect running e.

ades.—In order to obtain the most perfect type of blade, each ı machined out of the solid bar, each curve being given by a r or matrix fitted to the machine. In this way each blade is ly the same as any other, and exactly the same weight. Two ods of fixing the blades to the wheels are adopted. These are rated as (*a*) and (*b*), No. 50. The method shown at (*a*) is always ted for the velocity wheel, as it is naturally more suitable for this that shown at (*b*). The "Straddle" formation (*b*) is secured by ns of rivets (in double shear); while (*a*) is a dovetail, the portion he blade accurately fitting the groove formed in the periphery of wheel, which is provided with two "ports" to allow of the blades ıg put into position. Each blade is finely finished and polished, educe skin-friction to a minimum.

A general view of the blade wheels assembled on a shaft is shown (*d*), the velocity wheel at the extreme left carrying the two rows of pulse blades already described, followed by the single stage impulse eels. The balancing holes through these wheels are clearly shown this view, (*e*) is a photograph of a wheel being bladed; the disc has en cut to show the thickness of the metal.

The individual wheels are forced upon the shaft and driven by ther keys. The last wheel is secured by a nut with a fine thread. t (*d*) the complete rotor of a 2000-Kw. machine is shown. A cross ction (*c*) shows the wheels and diaphragms in place.

The axial clearance between the blades and stationary parts varies om $\frac{1}{8}$ in. in the smaller to $\frac{3}{16}$ in. in the larger sizes, and the radial earance between the tips of the revolving blades and the cylinders ı never less than $\frac{1}{4}$ in.

Main Shaft.—This is made of high quality steel, turned all over, nd stepped into diameter from the centre towards each end to take ıe blade wheels. Each wheel is provided with its own step and key. The stress in the shaft is very small, the latter being so designed that ıe critical speed is always far above the normal running speed.

Glands.—As the turbine has the advantage over the reciprocating ngine of being able to utilise the work in the steam to very low

pressures, it is most essential that a high vacuum be obtained, and this can only be acccomplished if air leakage is reduced to the lowest possible amount. A common cause of loss of vacuum with some turbines is due to leakage of air past the glands. No. 51 (*b*) shows

Vacuum. Atmosphere.

(a) Low Pressure Gland.

(b) High Pressure Gland.

No. 51.

the arrangement of gland at the high pressure end of the Westinghouse turbine and (*a*) that at the low pressure end. In each case the Westinghouse water gland is used which provides an absolutely air-tight seal between the inside and outside of the cylinder under all conditions of load and vacuum.

No. 52.—Steam Chest for Large Turbines.

The water gland consists of an impeller with open curved vanes rotating in a race. The centrifugal force exerted on the water tends to maintain an equal length of water leg on each side of the impeller, but the difference in pressure between the inside and outside of the cylinder forces a difference, *h*. It follows then that the centrifugal force on *h* is balanced by the pressure of the atmosphere due to the vacuum in the inside of the turbine casing. If the vacuum changes, the height *h* changes. Water is fed into this impeller race usually from a tank, placed about 20 ft. above the gland. As heat, due to friction, is generated in this water, the latter boils off (due to the lower pressure on this side, *i.e.*, about 28 in. vacuum, with boiling point about 100° F.) into the inside of the turbine casing, where it mixes with the steam from the turbine, and passes with it to the condenser.

The low pressure end has a gland of this type only (*a*), but the high pressure end is somewhat modified, as the pressure on the gland is considerably higher. This gland is shown at (*b*).

In this gland there is also, in addition to the water gland, a labyrinth gland, which is interposed between the high pressure chamber (where there is usually about 20 lbs. gauge on full load) and the water gland. The leakage past the labyrinth packing is very small indeed, and is led by a pipe into the exhaust space at the end of the cylinder. The water glands at each end of the cylinder are therefore under exactly the same pressure conditions. Important advantages of this gland are—that they are absolutely air-tight, the bearings are kept cool instead of being heated, no steam escapes into the engine room, and the conditions of operation are not upset by change of steam pressure or vacuum, as is the case with steam-packed glands.

Steam Chest.—Simplicity of design, and every precaution to prevent warping, under changing temperatures, and binding or deformation of the valves, have been the prior considerations in the steam chest shown in Nos. 52 and 53. A piston operated by oil under pressure controls both the governor valve and the overload valve through a relay. The steam pipe is coupled up at A, so that the stop valve B controls all steam taken into the turbine. This stop valve is of the balanced type, and is provided with a pilot valve for warming up. This standard design of stop valve has been found perfectly satisfactory after many years' experience. The strainer C, through which the steam next flows, consists of a perforated steel cylinder, the steam passing through the perforations to the main governor valve D. E is the automatic overload valve. Both valves are generally of the same type and size, and interchangeable, so that a spare valve is good for either position. They are of the well-known balanced double-seated engine type, fitted with self-aligning plates, so as to leave them perfectly free on their seats. They are very accessible, both D and E being held on their seats by a single keep-ring in the upper chamber. Steam from the valve D, admitted on the

No. 53.—Section through Steam Chest for Large Turbine.

No. 54.—Section through Steam Chest for Small Turbine.

throttle principle, flows direct into the turbine. The overload valve E is so arranged that it remains closed until the governor, having

No 56.—End View of High Pressure Westinghouse-Rateau Impulse Turbine.

 [To face page 557.

sufficiently opened D to enable the steam passing to carry full load, begins to lift H for further load. The valve rods pass through long frictionless glands. Pipes carry the steam from the valves D and E to the nozzle boxes or cylinder.

The Westinghouse steam chest for the smaller sizes (say up to 1000 kw.) is shown in No. 54. Steam enters at (A). The balanced stop valve (B) is provided with a pilot valve for warming up. In this

No. 55.—Turbine Governor.

case the overload valve on the cylinder is opened by hand. It is not possible for the steam chest or the cylinder to exert any pressure, due to expansion, the one on the other.

Governor Gear.—The Governor, No. 55, is specially designed to respond quickly to small variations in the speed of the turbine, while at the same time the parts are of a simple yet robust design. Practically no stress is thrown upon the pins or points, as the centrifugal force of the weights is opposed by the springs without the intervention of links or joints. The ends of the springs are threaded through thimbles, so allowing a nice adjustment to be made.

The direct connection by link gear between the governor and governor valves is shown in Nos. 56 and 57. The number of joints and simplicity of parts are worthy of note. A speeder spring enables the speed of the turbine to be changed to 5 per cent. above or below the normal whilst running.

Overspeed Safety Device.—All Westinghouse turbines are provided with an overspeed device which automatically prevents

any injury to either the turbine or the generator, in case an accident

Fig. 57.—Cross Section through Main Governor and Oil Pump.

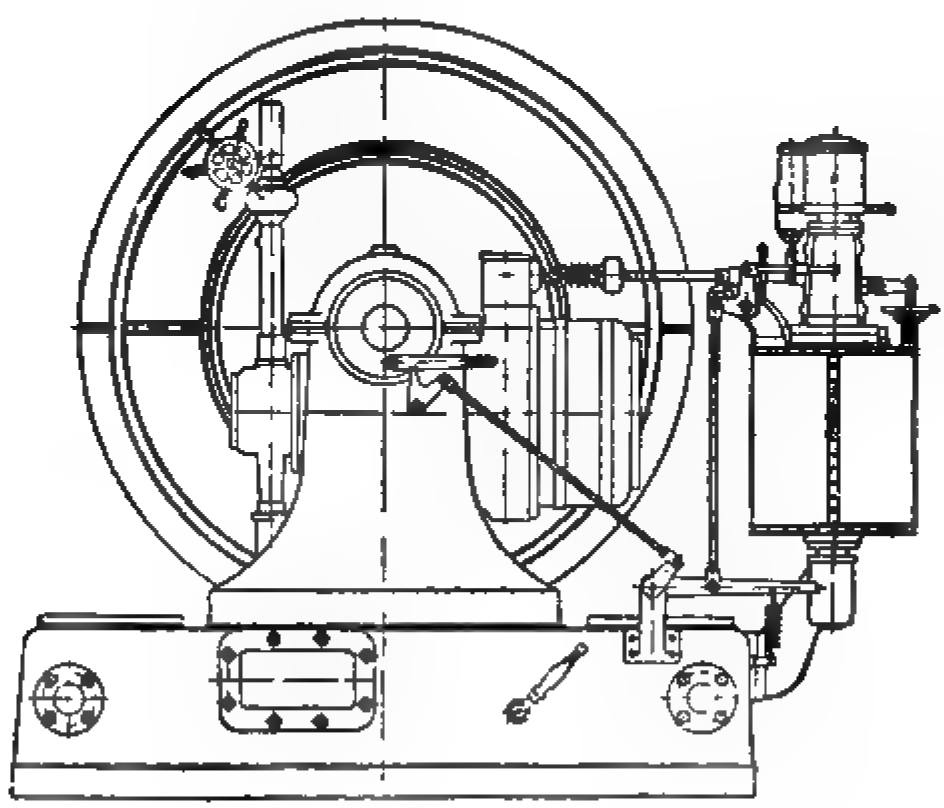
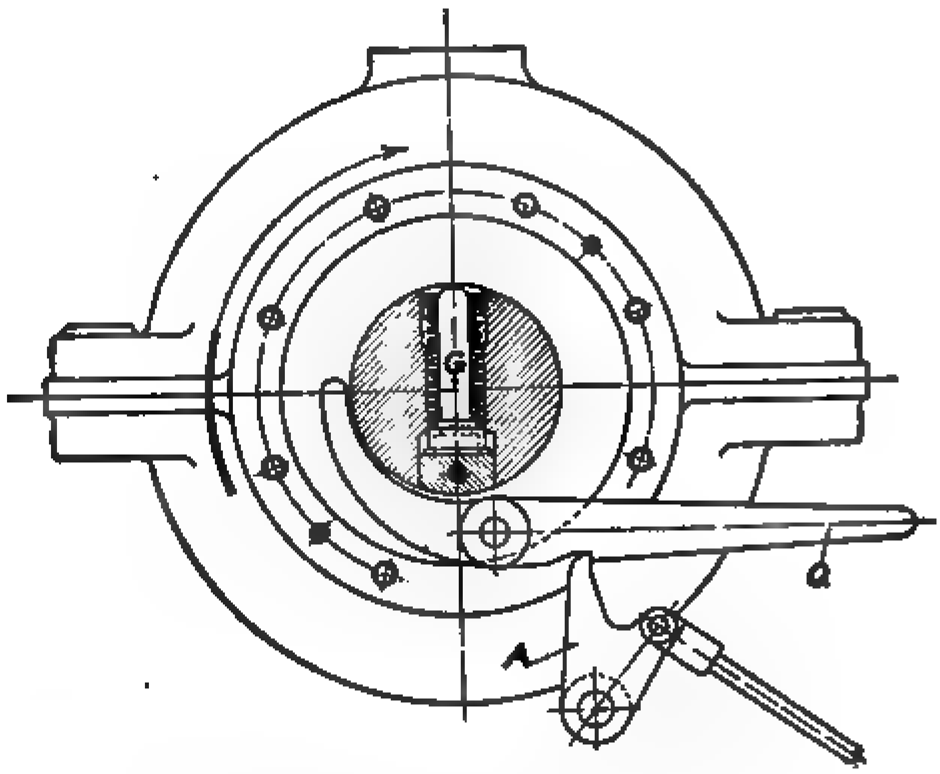


Fig. 58.—End View.

should have rendered the main governor valve inoperative, thus causing the turbine to exceed its normal speed. No. 59 shows a section through this safety governor, which is fitted on the end of

the main spindle. It is purposely placed on this first motion in preference to any other position, as intermediate gear might itself fail, and probably bring about serious results. It consists of an



No. 59.—Section through Emergency Governor, showing Plunger and Trip Gear.

unstable governor in which the weight, having its centre of gravity a small distance from the centre of rotation, is opposed by a spring of such a strength that the force exerted by it is less than the increase

No. 60.—Bearing.

in centrifugal force due to the weight moving further from the centre of rotation. Since this force is cumulative, immediately the weight begins to move, it flies out to its limit. This brings the rounded end of the weight into contact with the trigger slipping the catch A, so closing immediately the automatic stop valve in the steam chest

already described (see Nos. 53 and 54). The arrangement of levers through which this is accomplished is clearly shown in No. 58.

If desired, the turbine can also be shut down by hand, by merely lifting the handle Q shown in No. 59. It is desirable that this

No. 61.—Arrangement of Oiling System of High Pressure Turbine

should be done frequently, if not always, when shutting down. To reset the gear it is only necessary to lift the lever shown below the steam chest in No. 58.

Lubrication.—The lubrication of the complete set is carried out automatically, the oil being delivered under pressure to all the main bearings and to the various parts of the governor. The small oil pump shown in the bedplate in No. 57 is normally drowned, and draws its oil through a strainer. It then delivers, through the cooler,

to the various bearings, from whence it returns by gravity again to the oil tank. The oil pump is of the geared rotary type, so simple as to be practically free from the possibility of breakdown, and the strainer is fitted in such a manner that it can be withdrawn for cleaning purposes while the turbine is running; a small wing type hand pump drawing through a separate small strainer is used until the main strainer is replaced. This auxiliary oil pump is used also for priming the piping and bearings prior to starting up.

The oil pump fitted in the oil tank at the end of the bedplate is operated by means of a vertical shaft and bevel gearing. This

No. 62.—Thrust Block.

shaft is extended upwards, and drives direct a tachometer registering the speed of the turbine. This tachometer is immediately in front of the driver when he is standing at the steam chest, and paralleling the machine.

No. 61 shows diagrammatically the arrangement of the complete oiling system of a high pressure turbine equipped with oil relay operated valve gear.

Bearings.—The main bearings, No. 60, usually made of cast iron, are in halves and lined with high quality white metal. At each side and on the top and bottom a pad of steel is placed, under which is an assortment of liners of varying thickness, thus enabling the main spindle to be moved to a thousandth of an inch in any direction for

lining up, or to take up wear. Special straddle gauges are supplied, by means of which wear on the bearings can be ascertained at any time in a few minutes. The pads referred to are adapted to give the bearings spherical seating in their housings.

Thrust Block.—There is really no thrust on the turbine shaft, but some anchoring device is necessary to register properly the position of the shaft in relation to the cylinder, and since this takes the form of a thrust block, it has been designated as such. It is shown in No. 62, and consists of a cast-iron sleeve in halves. The thrust rings are lined with best quality white metal. They bear against, and mesh with, similar steel rings turned in the turbine shaft.

No. 63.—Coupling.

As inadvertently moving this thrust block might wreck the turbine, it is rendered fool-proof by locking it after adjustment in its correct position in the cylinder. Rings of the right thickness are inserted for this purpose at each side of the thrust collar. Oil, under pressure, is supplied to all the working faces.

Couplings.—After having considerable experience with both rigid and flexible type couplings the British Westinghouse Company has found that the former is preferable for high speed, and the latter for slow speed machines. For sets running at 2000 r.p.m. or more, the solid type coupling illustrated in No. 63 is fitted, and for sets operating at a lower speed a flexible coupling consisting of two "heads" and an outer sleeve. One of the heads is fitted to the generator shaft and the other to the turbine shaft. Both heads have teeth on the

outer rim, and the sleeve (which is sufficiently wide to cover both heads) has corresponding internal teeth. The sleeve is held in position by cover plates at each end ; by removing one of these the sleeve can be slid to one side, thus enabling either of the revolving elements to be removed without interfering with the other.

Low Pressure or Exhaust Turbines.

In general outline and appearance these turbines are similar to the high pressure type, but somewhat shorter ; they are also larger in the steam and exhaust pipes and valves. This is due to the less number of blade wheels and diaphragms, and to the quantity of steam dealt with being roughly twice that required for the same output from a high pressure turbine. No. 64 shows a longitudinal section through this type of turbine.

The constructional details are very similar to those of the high pressure turbine, with a few exceptions. The glands (of the Westinghouse water-sealed type) are the same at each end of the cylinder and the steam chest, and, instead of having an automatic overload valve, the turbine is provided with main stop valve and governor valve only. Overload valves are, as a rule, not fitted to exhaust turbines, but where circumstances render this particularly desirable, they can be provided. The above section does not show such a valve. For shutting off steam from the turbine in the event of over-running, a vacuum-breaking valve is provided instead of the emergency valve usually fitted to high pressure turbines. This valve is operated by a trip from the safety governor in the way already described for the emergency valve D as shown in No. 53, and it is further provided that, simultaneously, the governor valve shall be released and closed by its own weight. Under certain conditions, the governor valve is operated through an oil relay. This type of exhaust turbine is applicable to those cases where the quantity of exhaust steam to give the required output is restricted, and where the power to be obtained is large, and warrants the capital outlay. There are many cases where the gain to be effected by the installation of an exhaust turbine is small, unless the capital invested in it is small, and where there is more exhaust steam than sufficient, rendering economy of steam of secondary importance. To meet such cases, the British Westinghouse Company has developed a very simple turbine, the rotating element of which consists of a single velocity wheel with two rows of impulse blades. The cost of this turbine is much less than of that shown in No. 64. The details, however, are similar to those of the more expensive type.

With exhaust steam turbines, a reducing valve to reduce the steam from the boilers to about atmospheric pressure should always be provided, unless there is a perfectly continuous supply of exhaust steam to carry the load. A valve should also be provided to prevent a vacuum being formed in the exhaust pipes between the turbines and engines, as otherwise leakage of air into the system is almost certain to take place, with consequent loss in efficiency and output.

A good vacuum is of first consideration in the installation of an exhaust steam turbo-plant, since the pressure range, or more correctly

No. 64.—Westinghouse-Rateau Low Pressure Impulse Type Turbine.

speaking, the available heat range, is so greatly affected thereby. The lower the absolute pressure, the more rapidly are heat and energy surrendered; for instance, the energy given up by dropping from

28 in. vacuum to 29 in. vacuum (1 lb. absolute to 0·49 lb. absolute) is more than that given up in dropping from 27 in. vacuum to 28 in. vacuum (1·48 lb. absolute to 1 lb. absolute). A reciprocating engine is not able to take advantage of these very low pressures, but a turbine can do so with great efficiency.

Mixed Pressure Turbines.

In No. 65 we have the mixed pressure turbine, which is merely a non-condensing and pure exhaust turbine combined in one casing. A special arrangement of valves is necessary, as it is essential that the exhaust steam should always be used when available, and when not available the high pressure steam must be automatically and instantaneously admitted as the low pressure steam admission valve is simultaneously closed. Immediately the low pressure supply is again available the reverse operation takes place. The mixed pressure turbine is a particularly flexible unit, as generally it is capable of taking—(1) any load up to heavy overloads on high pressure steam, (2) full load on low pressure steam, (3) any load up to heavy overloads on mixed high and low pressure steam. With the exception that this is a slightly more complicated turbine both as regards blading and valve gear, and the steam consumption is slightly worse than a pure exhaust turbine, there is little doubt but that it approaches very nearly the ideal design for industrial works. In general design there is little to indicate any difference from the high pressure turbine. It will be noted, however, that a space is provided in the cylinder, and between the blade wheels on the shaft, for the introduction of exhaust steam. The portion of the blading to the left deals with the expansion of the steam from boiler to atmospheric pressure, and that to the right from atmospheric to condenser pressure. The most marked difference from the high pressure turbine lies in the steam chest and valve gear. The regulating valves for both high and low pressure steam admission are operated by power from a separate oil pressure system, but directly controlled by the main governor through relay valves.

This is shown in No. 66. The low and high pressure inlets from the steam chest to the cylinder are clearly indicated, and at the bottom of the steam chest on either side are shown the high and low pressure facings to which the pipes are connected. The governor valves for both low and high pressure steam operate perfectly automatically under any conditions of steam supply or load, the low pressure steam always being given preference over the high pressure, so that under all conditions the turbine operates with maximum economy.

There is absolutely no change in speed as the valves change over from high to low pressure steam operation or *vice versa*. No additional valve is necessary to prevent a vacuum being formed in the exhaust supply pipe. This ensures the least possible pressure drop, or wire drawing, of the exhaust steam, so that the heat value of the exhaust steam is fully utilised.

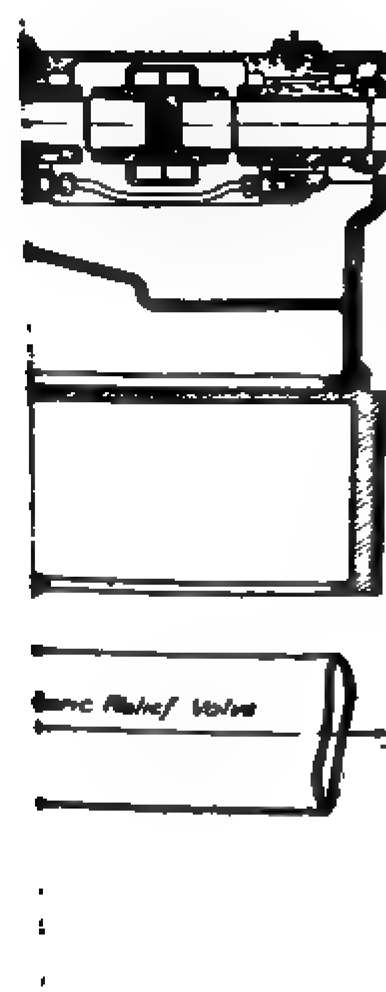
Back Pressure and Reducing Turbines.

These turbines are eminently suitable for those working conditions where steam, at a pressure which cannot be described as either boiler

No. 65.—Longitudinal Section through Westinghouse-Rateau Mixed Pressure Turbine.

or exhaust pressure, is used for some process in manufacturing, and where, at the same time, electrical energy is required. Boilers do not operate economically except at high pressures, so that in the event of

No. 66.—View of Mixed Pressure Steam Chest.



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No.

*'The Marine Steam Turbine.'

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a considerable quantity of steam being required at say 20 lbs. pressure above atmosphere, the most economical method of operation is as follows—boilers working at say 200 lbs. pressure, supply steam to a turbine where the energy between 200 lbs. and 20 lbs. is converted into useful work. If, instead of a turbine, a reducing valve were used the steam would be superheated when delivered to the 20 lbs. steam main, which superheat would most probably be lost in radiation very quickly, as the rate of radiation depends entirely upon the difference in temperature of the hot pipe and the surrounding cold air. Superheat in the steam might also ruin the product for which the steam would be required, if it were not lost by radiation. The third alterna-

No. 69.—Section through Small Turbine.

tive is to evaporate water at 20 lbs., which means very uneconomical boilers.

The back pressure turbine is applicable where the amount of steam required for heating is equal to, or more than, the amount required for power purposes in the turbine. For cases where the amount is less, the reducing turbine, which has a few additional Rateau elements for dealing with the surplus steam, comes into consideration, as the surplus can be used in the exhaust portion as economically as in a high pressure turbine. No. 68 shows a section through a back pressure turbine.

Small Turbines.

To meet the demand for electrical units of 25 to 200 Kw. output,

the British Westinghouse Company has designed and placed upon the market a simple small turbine, which can compete in price and efficiency with the high speed steam or gas engine. This type of turbine is shown in No. 69, and, as will be seen, the rotating element consists of a single velocity wheel. This wheel carries two or three rows of moving blades depending on the steam conditions and the available heat drop.

The machine is extremely light and compact, requires practically no attention, and finds great favour where first cost is of more importance than economy. For ship work or week-end factory lighting, as well as for driving high-lift pumps and other similar apparatus, the machine is particularly useful.

SECTION XIII.

SHAFT HORSE POWER, Etc.

THE Hopkinson-Thring Torsion Meter has been designed for determining, in a simple and accurate manner, the H.P. transmitted by turbine and other shafts.

The principle of the apparatus designed by Professor Hopkinson and Mr Thring is a differential one, and consists in the observation of the twist between two adjacent points on the shaft by means of two beams of light projected on to a scale from a fixed and a movable mirror. The beam projected on the scale by the fixed mirror is taken as the zero point, whilst the beam projected by the movable mirror indicates the amount of torque on the shaft. Both mirrors rotate with the shaft, but even at moderate speeds the reflections appear as continuous lines of light across the scale, and there is therefore no difficulty in taking readings.

The torsion meter is shown in No. 1, mounted complete on a shaft, and the scale box in No. 2, whilst a diagrammatic arrangement of the complete apparatus is shown in end elevation and plan in No. 3. A collar R, clamped to the shaft of which the torque has to be measured, is provided with a flange projecting at right angles to the shaft and an extension.

A sleeve T (No. 3), provided with a similar flange and extension at one end, is clamped at its further end on to the shaft in such a manner that its flange is close to that on the collar R, whilst its extension overlaps that of the collar R, on which it is supported to keep it concentric. Both the collar and sleeve are quite rigid, and it is therefore obvious that when the shaft is twisted by the transmission of power, the flange on the sleeve T will move relatively to that on the collar R, the movement being equal to that between the two parts of the shaft on which these fittings are clamped. This movement is made visible by one or more systems of torque mirrors mounted between the two flanges, which reflect a beam of light, projected from a lamp, on to a scale divided in a suitable manner on ground glass.

Each system of torque mirrors consists of a mounting, pivoted top and bottom on one or other of the flanges in which two mirrors are arranged back to back. This mounting is provided with an arm, the end of which is connected by a flat spring to an adjustable stop on the other flange. Any relative movement of the two flanges will turn

the torque mirror, and thereby cause the beam of light to move on the scale, the deflection produced being directly proportional to the torque applied to the shaft. Hence, if the rigidity of the material and the

No. 1.—Torsion Meter Mounted Complete on a Shaft.

number of revolutions per minute are known, the H.P. transmitted can be arrived at immediately.

With the arrangement described, a reflection will be received from each mirror at every half revolution of the shaft; but where the torque varies during a revolution (as with reciprocating engines), a

No. 2.—Scale Box with Lamp, for Torsion Meter.

second system of mirrors may be arranged at right angles to the first system, so that four readings can be taken during one revolution; or, if two scales are used, eight readings can be taken.

No. 3 shows how the beam of light reflected by the mirror when in its highest position passes through the upper part of the scale; while the second reflection will occur when the mirror is in the position

occupied by the zero mirror, the beam of light passing through the lower part of the scale. The position of the torque mirror in No. 3 is such that if the reflected beam strikes the scale to the right of the zero line, when the shaft has made a further half revolution, the reflected beam from the other mirror will strike the scale to the left of the zero line. Obviously the deflection on both sides should be equal.

The fixed mirror is attached to one of the flanges (in No. 3 to the flange of the sleeve T). This must be adjusted so that the beam of light reflected from it is received at the same point on the scale as those from the movable mirrors when there is no torque on the shaft. To facilitate the erection and adjustment of the apparatus, the box containing the scale and carrying the lamp is fitted with trunnions, so that it can be inclined as required.

If the position of the apparatus becomes altered relatively to the scale owing to the warming up of the shaft or from other causes, this is indicated immediately to the observer by an alteration in the position of the zero as reflected by the fixed mirror. Hence, the scale zero may be adjusted, if desired, by moving the scale so that its zero coincides with the reflection from the fixed mirror, but this is not necessary to obtain a correct result, since the mean of the two readings will be the same. It will readily be understood that a movement of the torque mirrors can only occur through a relative movement of the two flanges, so that vibration of the shaft or of the ship will not influence the readings in any way.

The constant of the instrument, viz., the factor which, when multiplied or divided into the product of the torsion meter reading and the revolutions, gives the H.P., may be calculated within 2 or 3 per cent., if the section of shaft within the instrument is uniform. But we recommend a direct calibration of the shaft with the instrument in position before the former is put into the ship. This is easily effected by applying a known twisting couple. It is no inconsiderable advantage of this instrument that a direct calibration is established between the torsion meter deflection and the torque on the shaft.

Calibration of the Hopkinson-Thring Torsion Meter by a Torsional Test of the Shaft.

The shaft to be tested should be bolted firmly at one end to prevent it turning, and provision made at the other end for applying the load by means of a couple, to obviate any unnecessary friction on the supporting bearings. The shaft should be very free in the supporting bearings, and should be vibrated by means of blows from hammers on the application of each increment of load and before taking the readings.

When received from the makers, the part of the apparatus which is to be mounted on the shaft is bolted together as it will appear when clamped on to the shaft, with the mirrors in position.

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7. Tighten the pinching screws firmly into the shaft, and lock by means of the nuts, and remove the four distance pieces and bolts N holding the flanges together.

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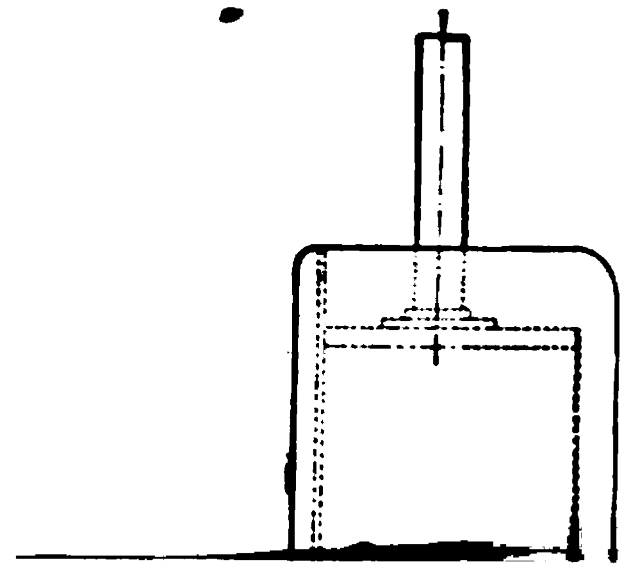

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When received from the makers, the part of the apparatus which is to be mounted on the shaft is bolted together as it will appear when clamped on to the shaft, with the mirrors in position.

It is most important that the directions here given should be carried out in the order stated, to ensure that the meter shall be clamped to the shaft without undue stresses being set up in the castings, which would be liable to cause errors in reading.

The operation of mounting should be proceeded with as follows:—

1. Remove the sheet-steel casing A (No. 3) covering the space between the two flanges so as to be able to get at the bolts holding together the two halves of the shorter collar casting and also the mirrors.

2. Unless the meter can be handled easily and without liability of damage to the mirror fittings, the torque mirror-mounting C should be removed from its frame by first taking out the capstan-headed screw D, and then depressing the spring E forming the lower pivot bearing, when the mirrors in their mounting with the flat spring F attached can be taken away.

3. Remove the bolts holding the two halves of the meter together, place the latter in the position on the shaft which it will occupy when used in the ship, and bolt lightly together so that it can be turned on the shaft if required, to enable the reflections from the mirror to be received on the scale.

NOTE.—The washers which will be found between the lugs of the short casting and at one end of the long casting must be left out when clamping to the shaft, but should always be replaced when bolting the two halves of the meter together, if away from the shaft.

The bolts must be put through the lugs in the same manner as found, *i.e.*, those on one side in the opposite direction to those on the other.

4. If the torque mirrors have been removed previously, they should now be replaced. Ascertain that the tightening up of the capstan-headed screw D has not strained the connecting spring F by loosening the nuts L, and see that the spring F is quite free on the end of the mirror arm. This being correct, tighten up the nuts L.

5. The scale box should be placed in position at the correct distance from the shaft, squarely and with the centre approximately opposite the centre line between the two flanges. If necessary, turn the meter on the shaft until the reflections of the lamp from one of the torque mirrors C and the zero mirror H are received on the glass scale, and adjust the lamp and lamp-holder until the reflection of only one limb of the filament is seen, and that squarely across the scale.

6. Clamp the meter tightly on the shaft, and ascertain that the distance from the torque mirror to the scale is correct, and fix the scale box. Care should be taken to tighten all the bolts equally, and for this reason they should be tightened gradually by small amounts at a time on each bolt.

7. Tighten the pinching screws firmly into the shaft, and lock by means of the nuts, and remove the four distance pieces and bolts N holding the flanges together.

8. Adjust the glass scale G endwise in its groove, so that the reflection from the zero mirror falls on the zero of the scale, and adjust the torque mirror so that its reflection is superimposed on the zero reflection.

The meter is now ready for recording the twist taking place in the shaft on the application of torque.

NOTE.—Before adjusting the reflection to zero, the shaft should be vibrated by means of blows from a hammer, to ensure that it is free in its bearings.

The readings should be taken as between the reflection from the moving, or torque mirror, and the fixed, or zero mirror, and not necessarily from the zero of the scale. This will eliminate any error which would arise from endwise movement of the shaft or scale box.

The readings on the scale being plotted against the torque (in ft.-lbs.) which produces them, the constant of the instrument can be found by the following method :—

As the expression for the power (S.H.P.)

$$= \frac{\text{Scale Readings}}{\text{Constant}} \times \text{Revs. per minute, expressed in H.P.,}$$

$$\text{and since the power (S.H.P.)} = \frac{T\,2\pi}{33000} \times \text{Revs. per minute,}$$

$$\therefore \text{Constant} = \frac{\text{Scale Readings}}{T\,2\pi} \times 33000.$$

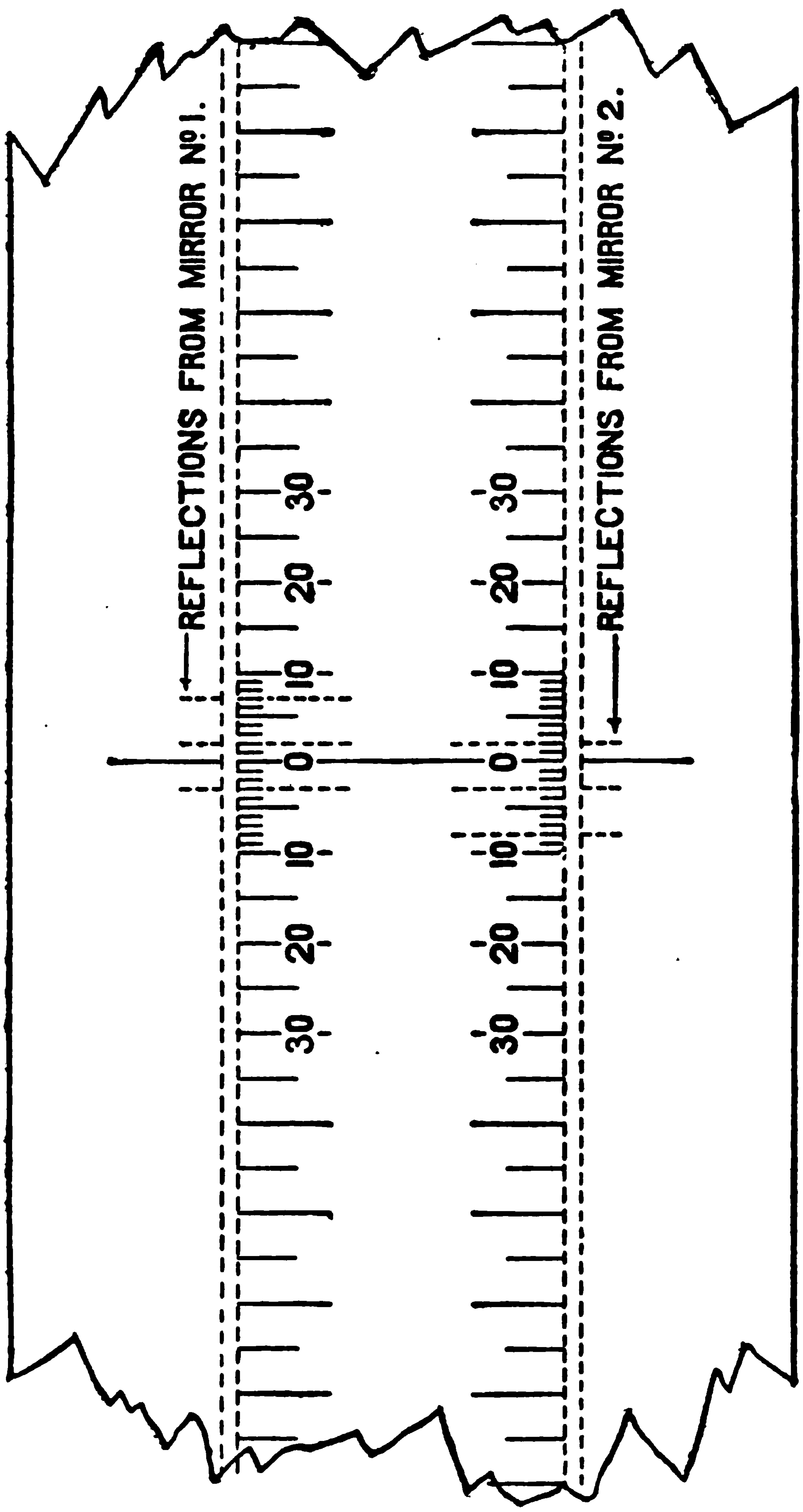
For calculating the modulus of rigidity of the shaft, or for the purpose of the formulæ for the "Determination of H.P. Constants," the length of the mirror arm will be found engraved on the mirror mounting. The other particulars and dimensions must be taken from the drawings of the instrument and shaft.

Setting up and Adjusting the Meter on the Shaft in the Vessel.

The erection of the meter on the shaft is exactly the same as the previous instructions given under the heading "Calibration of the Hopkinson-Thring Torsion Meter," but in this case, after the clamping bolts have been tightened up, the split pins must be put through the ends of the bolts.

For setting the mirrors correctly, the following operations should be carried out :—

A. The scale box should be placed in position at the correct distance from the shaft, with its centre opposite the mirrors. The position for this is found by turning the shaft until the reflection of the lamp from the zero mirror H is thrown on to the glass scale, and by adjusting the lamp and lamp-holder with its shield, which is made to rotate until the reflection of one limb only of the filament is seen, and that squarely across the two scales. Adjust the scale box endwise until the white lines provided on the front edges are in line with the reflection. Look through the scale towards the meter, and


REFLECTIONS FROM MIRROR Nº 1.
REFLECTIONS FROM MIRROR Nº 2.
30
20
10
0
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30
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No. 4

turn the lamp box on its trunnions, if necessary, so that the mirror can be seen through the clear glass strip of the glass scale, and clamp the trunnions of the scale box.

B. Turn the shaft until one of the torque mirrors reflects on to the scale, and note the reading on the scale. Continue turning the shaft for a short distance in the same direction, and reverse until the same torque mirror as before reflects on to the scale, and note the reading again. Repeat these operations two or three times to obtain an average reading, as it may be found that the friction is not constant with the slow motion of turning.

The distance on the scale between the two readings obtained thus represents the twist due to the friction of the shaft when going ahead, plus that when going astern. The zero, or the position of the reflection on the scale, if there were no torque on the shaft for the mirror used, is therefore midway between the readings.

C. Turn the shaft until the other torque mirror reflects on to the scale, take readings in the same manner as for the first mirror, and ascertain the zero position as before.

D. The mirrors now require to be adjusted so that the two zeros are superimposed. To illustrate the adjustment of the zeros we give a diagram (No. 4), showing the readings on the scale such as might be obtained by the operations described in paragraphs B and C.

From the diagram it will be seen that the zero of mirror No. 1 is two divisions to the right of the present scale zero, and that of mirror No. 2 three divisions to the left. These two zeros can be made to superimpose, the point on the scale being the mean position, which is half a division to the left of the scale zero.

With one of the torque mirrors, say No. 2, reflecting on the scale, move the scale one half division to the left, and adjust the torque mirror by turning the nut K, so that the reflection from the mirror, which was eight divisions to the left, is now five divisions to the left, when it will be found that the other reading from the same mirror will be five divisions to the right. The two mirrors being mounted together, the adjustment of one also adjusts the other, and it will be found, therefore, that the readings from mirror No. 1 will now be five divisions on either side of zero.

The reading of five divisions represents the amount of twist due to the friction of the bearings.

After adjusting the mirror, tighten up the screw Q.

E. The fixed, or zero mirror, now requires to be adjusted so that it reflects on to the scale zero, which can be done, first by turning the shaft until the mirror reflects on to the scale, and then by turning the screw at the back of the fitting to bring the reflection to the zero line of the scale.

After adjusting all the mirrors, and examining all the nuts and screws to see that they are tight, replace the sheet steel covers between the flanges.

MICHELL THRUST BLOCK.

In this, the latest type of thrust block, the pressure is taken up by a set (5 to 8) of loose "pads" fitted in the panels of a containing frame ring, or cage, which forms the block. The pads, which are segmental in shape, are adjusted to allow of a slight rock or "wobble" by being pivoted at or near their centre, and the effect of this slight accommodating motion is to allow a liberal oil film clearance between the bearing surfaces of the block when in motion. A steel button with a hard rounded point serves as a pivot on which the pads cant over when acted on by the thrust pressure, and oil service.

The shaft contains a single thrust collar, one side serving for ahead thrust pressure and the other for astern thrust pressure, and, as the collar revolves, the oil is forced up continuously to the surface of the pads or blocks, the result of which is, that the pads will take a pressure per square inch of about ten times more than that of the ordinary type thrust block without distress (500 lbs. □″ in place of 50 lbs. □″). The segmental pads are usually of gun metal, with bearing surfaces of white metal.

The bearing blocks are adjusted to the correct setting by means of flat rings (see Sketch); also by a spherical seated ring of steel with hardened steel contact points to bear up against the back of the pads.

With geared down turbines the steam pressure on the blades, acting in one direction only, requires to be balanced, so that a Michell thrust is fitted on the turbine shaft in addition to the heavier type of Michell thrust fitted on the main or driving shaft to take up the actual propeller thrust.

The main shaft thrust is made in exact duplicate on each side of the shaft collar to take up ahead or astern running as required.

If the H.P. turbine is for ahead running only, and no H.P. astern turbine is fitted, then the thrust is single way only, a plain white metal faced ring being sufficient to take up any reverse thrust pressure likely to come on the block in this case (Sketch 13).

As, however, the L.P. turbine and reverse turbine are on the same shaft, then an astern thrust is also necessary, so that the Michell thrust is again made duplicate on either side of the shaft collar as in the case of the main shaft thrust block.

* "The principle of the Michell bearing is that of *pressure oil film lubrication*. That is to say, that the rubbing surfaces are entirely separated by a high pressure oil film. There is no metallic contact, consequently there is an enormous reduction in friction; in fact at moderate speeds it is a negligible factor.

"The pressure of oil film is generated automatically by the rotation of the shaft, no auxiliary pump being necessary.

"The fact that under favourable conditions fluid pressure is generated in the oil film of a journal bearing has been known for

* Reprinted from "The Michell Bearing Book," by Mr H. T. Newbigin, Esq., A.M.I.C.E.

many years, but it is only recently that the theory of lubrication has been sufficiently well understood to enable similar results to be obtained in thrust bearings.

"One of the essential conditions for the maintenance of a pressure oil film between two lubricated surfaces is, that the surfaces must have a slight inclination to each other, so that the opening at which the oil enters is greater than that at which it leaves the brass. In journal bearings this takes the form of a curved wedge, due to a slight shifting of the centre of the shaft in relation to the centre of the brass, but in a thrust bearing, as ordinarily made, the surfaces are necessarily parallel.

"The coefficient of friction $\left(\frac{F}{W}\right)$ is at least ·03, and if the pressure exceeds about 60 lbs. per square inch the surfaces seize. These are the conditions which obtain in an ordinary thrust bearing.

"But, if the block is free to assume a slight inclination to the surface, it then rides on the oil film and there is no metallic contact.

"The coefficient of friction $\left(\frac{F}{W}\right)$ falls to ·0015 or less, and the pressure per square inch may be safely raised to 500 or 600 lbs, without danger of metallic contact. This is the condition which obtains in a Michell thrust bearing, and such bearings have been run cool under a pressure of *five tons per square inch*, at which the white metal surfaces of blocks began to squeeze out like butter, thus showing the enormous loads that the oil film will carry in bearings made on the Michell system.

"The best shape in which to make the blocks, and the most favourable position at which to pivot them, have been worked out theoretically and proved by experiment.

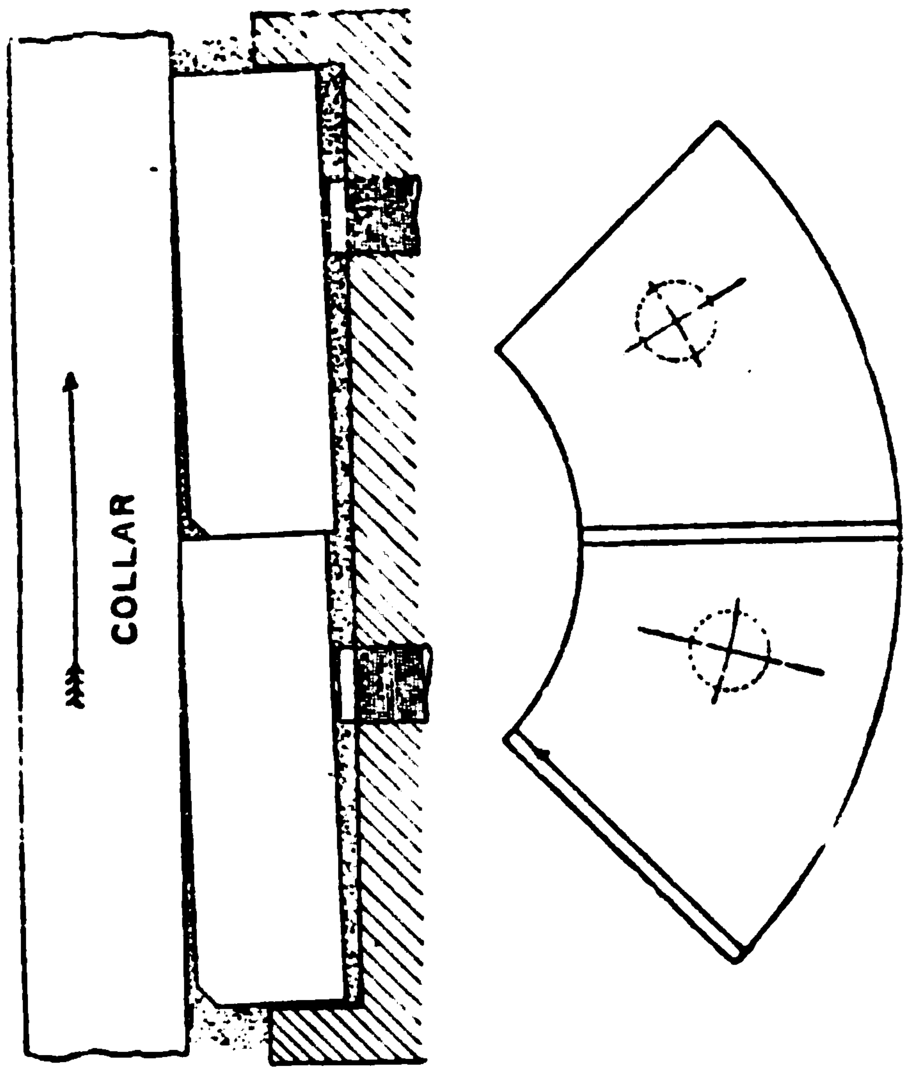
"The thickness of the oil film is very small, being sometimes less than one-thousandth of an inch; it is therefore very important to have the surfaces properly scraped up, and on no account should any oil grooves be cut in them, as the oil pressure would thereby be allowed to escape.

Mechanical Details in Thrust Bearings.

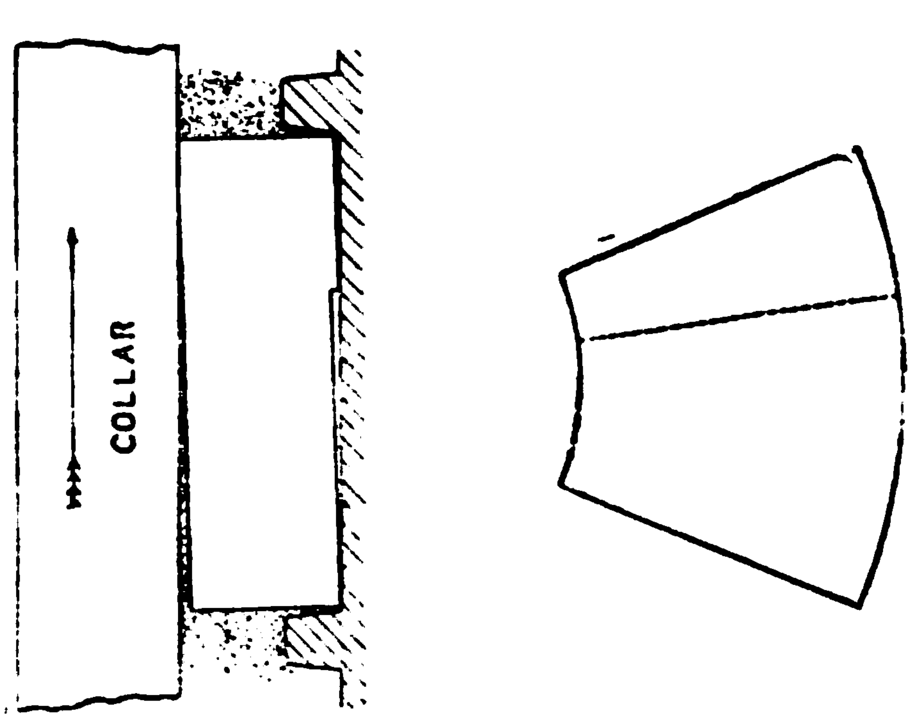
"In the ordinary type of thrust bearing, in order to bring the pressure per square inch down to one that the oil film will carry, a number of collars are provided on the shaft, which bear against a corresponding number of annular surfaces, but in the Michell thrust bearing, as very much higher pressures can be carried, it is not necessary to provide more than one collar. The surface against which this collar bears is, however, not continuous as in the ordinary thrust bearing, but is divided into a number of segments, each pivoted at its back so that it is free to assume a slight inclination to the plane of rotation. This pivoting provides automatically the essential condition for the formation of the pressure oil film already described.

"The special feature of the Michell thrust bearing is, then, the

subdivision of the stationary surface into a number of pivoted segments. Each of these segments is made to pivot either on a line, or on a point, both of which are quite satisfactory in practice, in spite of any slight crushing that may sometimes occur along the line or at the point of contact with the supporting member.



No. 6.
Illustrating Point Pivoting in a Michell Thrust Block.



No. 5.
Illustrating Line Pivoting in a Michell Thrust.

"In Nos. 5 and 6 the two methods of pivoting the segmental blocks are shown. In the first the blocks are bevelled off or stepped on their back surfaces, so as to allow them to tip on a radial line. In the second they are pivoted on the rounded end of a pin or screw, and free to tip in any direction.

"The blocks are prevented from moving circumferentially by means of stops. These may be placed at each end of each block as in No. 5, or at the ends of a series of blocks as in No. 6.

"The amount of tilt taken by the blocks when at work is of course enormously exaggerated in the above figures.

"The use of pivoted blocks is common to all Michell bearings, but in the thrust bearings it is necessary to provide a means of distributing the load equally amongst them.

"(1) By mounting the series of blocks on a spherical seat as shown in No. 10.

This gives automatic adjustment.

"(2) By independent hand adjustment as shown in Nos. 7 and 8.

The Marine Thrust Bearing.

"The most important bearing in a mechanically propelled vessel is undoubtedly the main thrust bearing, as against it the whole force moving the ship must press.

"In the ordinary multi-collar type the friction is *proportional to the thrust*, consequently when the thrust increases, the heat generated also increases. If, therefore, the bearing is adjusted to suit the thrust under normal conditions, it will run hotter if these conditions are altered, say, owing to the vessel driving against a strong head wind, or towing another. The effect of this is to cause unequal expansion and throw more weight on to some collars than on to others. This again generates more heat, until a "hot thrust block," with all its possibilities of further trouble, may be and often is the result.

"In the Michell thrust bearing, not only is the friction *independent of the thrust*, but there is only one collar, consequently expansion troubles cannot occur.

"Although the ordinary type of thrust block has been in use for many years for reciprocating engines, it proved to be a complete failure when fitted to a geared turbine drive. And the opportune use of the Michell thrust bearing has been a most important factor in the success of this type of marine engine. It is also rapidly replacing the older pattern for reciprocating engines.

"In the multi-collar type of thrust bearing the thrust is usually taken on a number of horse-shoe shaped pieces, which transmit their load on two parallel side screws, by means of nuts threaded on them. There are four nuts to each shoe, so if there are ten of these shoes there must be forty nuts, all of which must be set with the greatest care.

"Nos. 7 and 8 show sectional side and end elevations of a Michell marine thrust bearing suitable for either a reciprocating or a geared turbine engine. In this there is only one collar on the shaft, and it bears against two inverted horse-shoe shaped surfaces (one for ahead and one for astern thrust). Each of these surfaces is sub-divided into six blocks or pads pivoted on the ends of a corresponding number of screws, which are tapped into and pass through the ends of the housing. So that in the equivalent Michell thrust bearing there are only twelve adjustments to be made, as against forty in the corresponding older pattern.

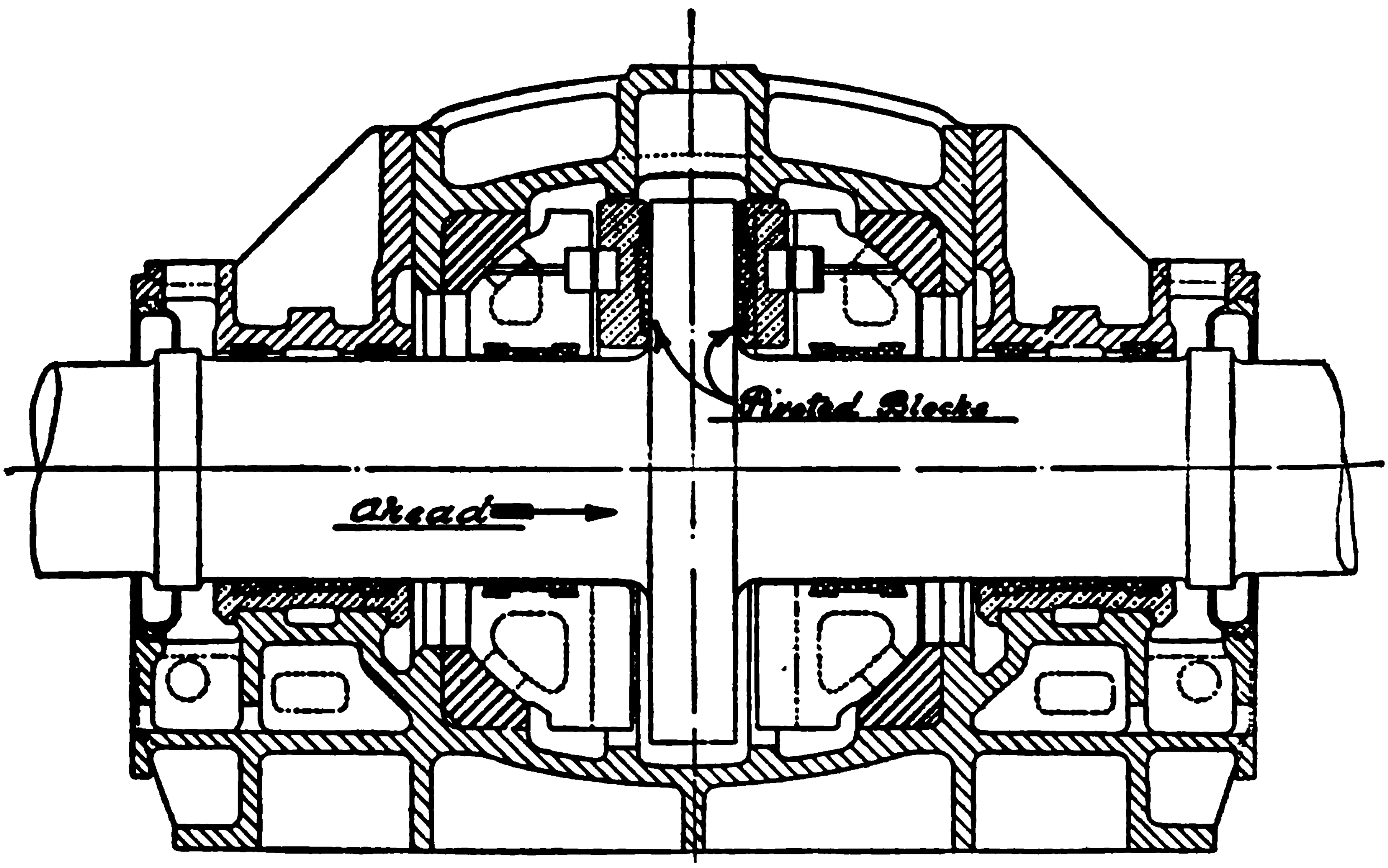
revolves is filled about half-full of o
lubricated. Keeps on the journal bea

cover complete the arrangement. In the case of large sizes, or of those running at a high speed, the housing is water-jacketed.

"A considerable number of this pattern are at work on both turbine and reciprocation engine-driven vessels.

"Some of $7\frac{1}{2}$ in. shaft dia., tested recently, gave the following result:—Load, 20,000 lbs., or 400 lbs. per square inch of thrust surface; speed, 460 revs. per min.; cooling water used, $\frac{1}{2}$ gal. per min.; coefficient of friction, about ·0015.

"No. 10 shows a section of another type of marine thrust bearing largely used in connection with geared turbine installations. The

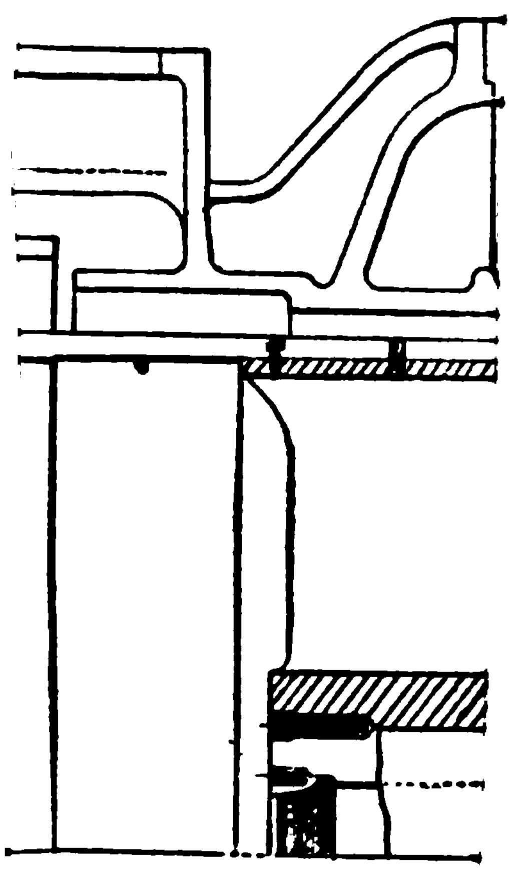


No. 10.—Illustrating an Enclosed Type of Marine Thrust Bearing.

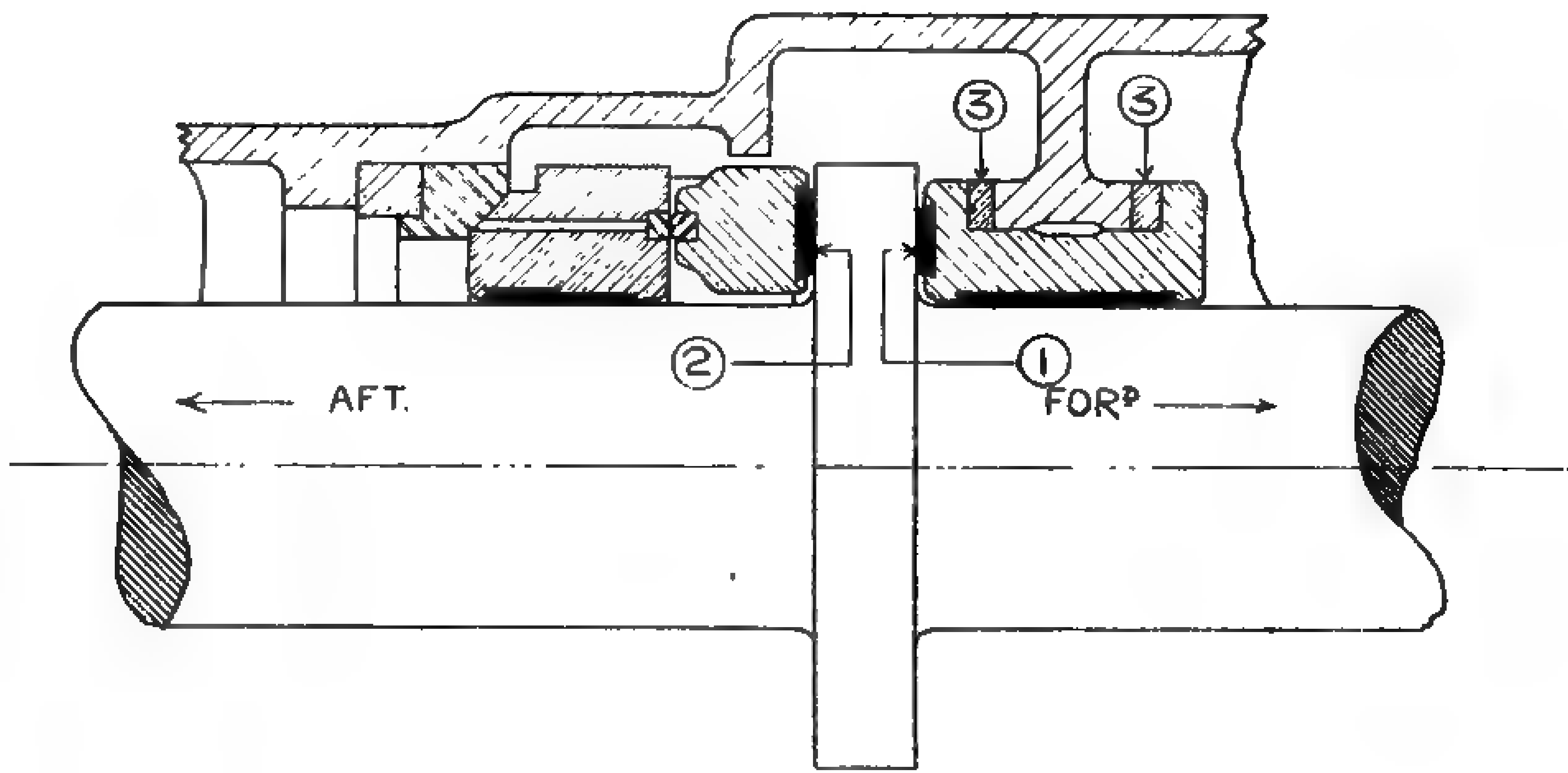
NOTE.—About $\frac{1}{16}$" running clearance is allowed between the ahead and astern faces.

shaft is carried in two ordinary journal bearings as in the last case, but it differs in other respects. The housing is made in halves, and the blocks, instead of being mounted for independent adjustment, are mounted in spherical seats so that they may adjust themselves automatically. This type requires to be connected to an oil circulating system; it is not self-contained as regards lubricating like the last example.

"The block may be either 'line pivoted' on their spherical seats, or 'point pivoted,' as shown."



The white dotted line shows where the white metal is usually washed away ($\frac{5}{1000}''$) near the trailing edge, with the object of allowing a free oil film flow across the block when tipped.



No. 13.- :k.

This type is fitted on H.P. steam pressure on the blading ac of the rotor forward through any of thin adjustment rings. as usually arranged. The longitudinal displacement :ld in position by a couple

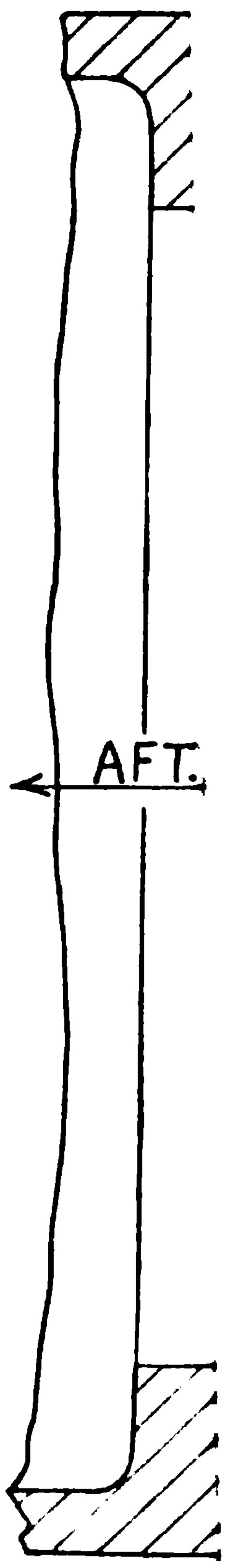
1. "Astern" surface. 2. Ahead surface. 3. Adjusting rings of astern side.

No. 6.—Michell Patent Single Collar Thrust Block.

This view shows cover removed, and one of the bearing pads or "kidneys" taken out of position.

No. 7.—Michell Patent Single Collar Thrust Block.

View showing cover removed and bearing pads in position, also spherical blocks and liners, etc.



The
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drum in

Macalpine Floating Frame Reduction Gear.

Although the rigid type gear frame has now been in extensive use for some years, and has given quite satisfactory result sin the majority of cases, it is quite obvious that a flexible type gear frame must necessarily be an improvement in the rigid type, as, if any change takes place in the longitudinal position of the driving pinions through wear or other local causes, the pressure on the teeth may easily rise to a very high figure, resulting in breakdown or damage to the working parts. With the Macalpine type—flexible or floating type gear—the inner stiff frame which carries the pinion bearings is free to

No. 15.—Construction of Alquist Flexible Reduction Gear Wheel.

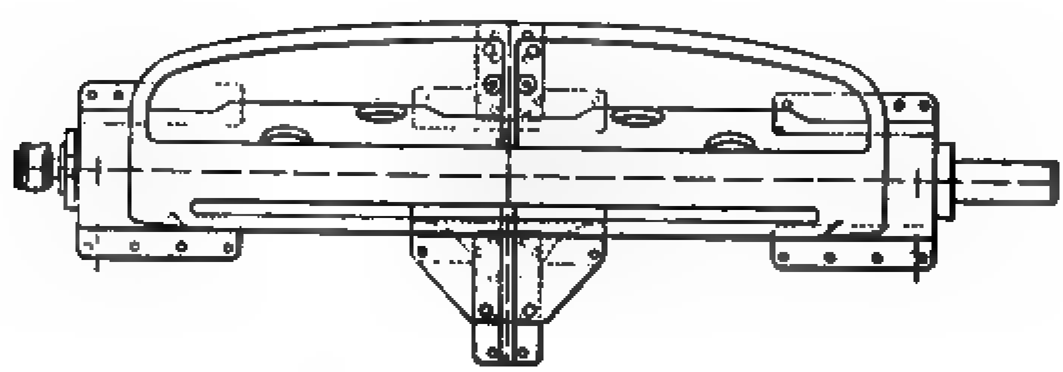
slight movement or tilt about the centre of the pinion length, and this automatically neutralises any change in pinion position (vertically or horizontally) and resultant teeth pressure. An attempt in this direction of flexibility has already been made in the case of the "Alquist" gear, which has been fitted by the General Electric Company of U.S.A. in many turbine-driven vessels, and has given satisfactory results (No. 15).

In the "Alquist" system the large gear wheel is built up in three or four separate discs, which are strung together on to the driving shaft, the hubs of each disc being in contact, as is also a narrow fitting strip rim on the outer periphery.

The wheel being in separate sections, in place of being solid, allows of slight corrective flexibility when the effects of deflection,

torsional yield, or other misalignment troubles to which the pinions are subject, take effect.

We now quote from the excellent paper entitled "The Design and Progress of Floating Frame Reduction Gear," by John H.



No. 16.—Macalpine Type Floating Frame Reduction Gear.
B shows I section flanged beam which allows of pinion flexibility or floating adjustment.

Macalpine, Esq., and read at the meeting of the Institution of Engineers and Shipbuilders in Scotland, on 19th March 1918, and which can be commended to the notice of those interested in the subject. Referring to rigid gears of the De Laval type, Mr Macalpine says :—

"For good economy de Laval's single-stage turbine must run very fast, and, therefore, the ratio of gear to pinion diameter must be large in order that the speed of rotation of the large gear may be sufficiently reduced for driving pumps, blowers, and so forth. Thus the pinion is made very small in diameter, so that the gear-wheel diameter may be kept within reasonable limits. To give sufficient tooth contact this small pinion must be made as long as practicable. This introduces two difficulties of great importance—(1) A slender pinion is liable to cross-bending between the bearings. (2) The torsional yield of the pinion becomes important. Both these distortions cause bad distribution of tooth pressure. The torsional error is in no way counteracted in the de Laval gear, and would, alone, limit the average pressure which can be applied per inch of tooth. It is sought to prevent cross-bending by placing the pinion—in all but turbines of small power—between the two gear wheels. It thus pushes up on one side and down on the other. If the powers transmitted by the gear wheels are equal there is no cross-bending of the pinion, but it is not possible to maintain this equality at all times, and then serious cross-bendings may occur. Thus this is never a safe way of eliminating the cross-bending error in any gear design.

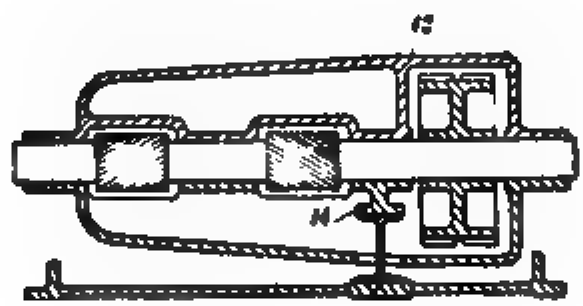
"More important than either of these, in any gear whatever which has all the pinion and gear bearings cast in one solid housing, the slightest warping of this casting, or other causes, will put the pinion out of line and cause highly localised and intensified tooth pressures, especially if the pinion is long. (Where there are two gear wheels, as here, this danger is intensified.) Even if set true at first, misalignment is practically sure to occur during the life of the gear, and the low power constants used are proof of this. These are, I believe, the points in the design which have confined the de Laval gear to small powers. . . ."

"As proof of the presence of these causes affecting very detrimentally the distribution of tooth pressure in these gears, I give the data of a 300 H.P. de Laval gear, which I refer to later in the text, and compare it with that of a floating-frame gear having the same diameter and revolutions per minute of pinion, and a power constant = 4.

	De Laval Gear.	Floating Frame Gear.
Horse-power - - - -	300	691
Pinion, pitch diameter, inches - -	2·542	2·542
" R.P.M. - - - -	10510	10510
" number of bearings - -	2	3
Length of each helix, inches - -	9·5	5
Helical angle - - - - -	45°	30°
Pitch-line speed - - - -	6994	6994
Pressure per inch of total face - -	37	326
Power-constant - - - - -	1·738	4·0

"In the de Laval gear, if equal power could certainly be taken from each large gear, annulling cross-bending, and if alignment could

Macalpine Floating Frame



No. 20.

10A. Stationary outer frame.
12. Floating frame of idler.
13. Flexible I section beam of idler, which is bolted at bottom to stationary frame 10A and at top to flange (14) of floating frame.
16. Floating frame of turbine pinions.
16A. Flexible I section beam of turbine pinions.

The angular movements of the flexible I section beams can be limited by the fitting of suitable struts as required.

No. 17.- Macalpine Type Floating Frame Reduction Gear.
(Partly in Section.)

This illustration shows the large experimental gear of the floating frame with the housing and frame partly broken away, the I. section flanged beams which allow for "floating" adjustment being seen near the centre of the flame.

No. 18.—Macalpine Type Floating Frame with Double Reduction Gearing.

Each floating frame is fitted with three bearings. The second reduction floating frame carries the large gear wheel of the first reduction gear and the pinion of the second reduction gear.

H.P. 3400	Small pinion revs.	= 3740	$3740 \div 450 = 8{\cdot}31$ Ratio.
	Large ,,	= 450	
	Propeller shaft revs.	= 72	$450 \div 72 = 6{\cdot}25$ Ratio.

1st reduction as 8·31 : 1
2nd ,, 6·25 : 1
Total reduction $= 8{\cdot}31 \times 6{\cdot}25 = 51{\cdot}9$ 1

No. 19.—M'Alpine Type Floating Frame Double Reduction Gear.

This shows another view of the same gear as described in No. 18, the upper halves of the four floating frames being removed.

certainly be maintained, the power constant should be very high, as the pinion, driving by both sides, gives a tooth face of 38 inches. Instead one finds a low power-constant used. By adding a third bearing to the pinion and shortening the helix, cross-bending and torsion is reduced to unimportance; the floating frame wholly annuls, as will be shown, the effects of misalignment; the smaller helical angle gives a stronger tooth—the action becomes so uniform that a large helical angle is no longer required for quietness; one large gear is entirely dispensed with; the high power-constant becomes quite safe; the tooth pressure per inch is raised nearly nine times; and the power is more than doubled.

"*Corollary*: In so far as this proves the de Laval gear to be liable to misalignment, it also proves every rigid gear to be liable. Hence the lower-power constants adopted for rigid gears."

Regarding the "power constant" referred to, Mr Macalpine says:—

"Perhaps the most important constant in connection with gearing is what I have called the 'power constant.' It is:—

$$\text{Power Constant (C)} = \frac{1000\ P}{D^3\ R}.$$

Where, P = H.P. transmitted,
,, R = Revs. per min. of pinion,
,, D = Diameter of pinion in inches.

"The factor 1000 is introduced to bring C to a value usually between unity and 10. . . ."

"From actual tests it can easily be proved that the floating frame will allow of a much higher power constant than them of the rigid type; therefore higher speed of pinions are permissible, and load on pinion teeth, which results in lighter and smaller gear for a given power transmission.

An examination of Table V., given in the presidential address of Mr Alex. Cleghorn, and which shows very clearly the comparison between the rigid and floating frames, brings out in strong relief the decided advantages of the latter in almost every essential point."

In marine practice double reduction gearing of the Macalpine type, with a 50 to 1 ratio, have been already fitted and run, the results obtained being just as was anticipated, and satisfactory in every possible way.

It will be observed that the bearings of the large gear wheels are solid with the outer case of the frame, while the pinion bearings are arranged on another and independent inner frame, supported on flanged beams, which by suitable adjustment possesses sufficient flexibility to allow of any slight change in longitudinal displacement of the pinions, and thus automatically allow of self-adjustment of tooth-bearing surface and pressure under all conditions of alignment.

Westinghouse-Leblanc Rotary Air Pump.

Rotary air pumps and "kinetic" air pumps are both of the "dry" air pump class, and are not intended to be used for the extraction of

the condensate, which requires independent handling by separate pumps.

Rotary or kinetic air pumps are employed to obtain the highest possible degree of vacuum as required in the case of turbine practice. To those already familiar with the action of the mercury pump employed in the manufacture of filament electric lamps, the action of the rotary air pumps will easily be recognised as similar, except that small plugs or pistons of water take the place of the little plugs of mercury, the air being caught and imprisoned in exactly the same

No. 21.—Westinghouse Leblanc Dry Air Pump.

manner between each pair of water pistons, and so extracted from the condenser.

The principle of the Leblanc air pump differs from all ejector types, which depend on friction for the entrainment of air.

The internal construction is shown in the sectional view, No. 20.

The sealing water enters the pump casing at branch A, passes through the fixed guide nozzle B to the impeller, which is a reversed Pelton wheel. The water is projected by the impeller into the chamber D in the direction E in a series of sheets, forming with the impeller as it enters the air chamber D a water packed radial bladed wheel, which scoops out the air as it rotates.

The air enters the air chamber D through inlet C.

These sheets, with the small pockets of air entrained between them, are driven with high velocity through converging and diverging cones E and F into the diffuser H, where the velocity is converted into pressure sufficient to overcome the external head due to the atmosphere; and when discharging under water, the corresponding head added to the atmospheric pressure.

No. 22.—Leblanc Ejector Air Pump.

In practice the efficiency of the Leblanc air pump increases as the vacuum becomes higher, owing to the absence of "waste clear-

No. 23.—Leblanc Ejector Air Pump.

ances" and pulsation in the flow of the air to the pump—defects inherent in the reciprocating pump.

The reciprocating pump generally extracts the air at the temperature and pressure in condenser, the air pressure being the condenser pressure less the vapour tension at the point of extraction,

With the Leblanc air pump the temperature of the air, and the tension of the vapour with which it is saturated, are reduced to that of

the sealing water, the volume being proportionately diminished, enabling the air pump to remove effectively from the condenser a proportionately increased weight of air per cubic feet displacement of pump.

* "The British Westinghouse Electric and Manufacturing Co. Ltd., Manchester, and the firm of Messrs The Mirrlees Watson Co. Ltd., Glasgow, manufacture an ejector air pump, under licence from the Societé Anonyme Westinghouse, Paris and Le Havre, which is another invention of Monsieur Maurice Leblanc. It is the outcome of many months of arduous research work, during which time innumerable difficulties were surmounted by the inventor, with the result that a really first-class ejector air pump has been evolved. Nos. 21 and 22, show the general arrangement of this apparatus. It will be noticed that the pump is arranged to work in two stages, and the steam is admitted to the second stage of the ejector by opening the stop valve C. Immediately C is opened steam fills the annular space behind the nozzle plate, and finds its way into the throats of the group of nozzles attached to this plate, it then passes along the steam pipe which supplies the first-stage nozzles, which are also attached to a nozzle plate. The supply of steam in this set of nozzles is controlled by the stop valve on the steam supply pipe. The pump is connected to the condenser at the branch D, which is the air inlet branch. At the entrance to each of the steam spaces fine wire gauze strainers are fitted to prevent any foreign matter, which may have primed over with the steam from the boilers, from entering the nozzles, thereby intercepting any stoppage in the nozzle throats, and consequently a loss of vacuum. These nozzles are efficiently locked to the nozzle plates. The mixture of air and steam is discharged at the mouth of the divergent cone Y, and led away to the boiler feed tank, so that the heat units contained in the operating steam can be reclaimed by heating the feed water.

"To start the pump to work it is only necessary to open up the steam valve C, and the vacuum will at once commence to increase in the condenser or other vessel to be evacuated. When the vacuum gauge becomes stationary the second stage steam inlet valve is opened up to bring the vacuum to a maximum. A very important feature in this pump is the absence of moving parts. The simplicity of the apparatus is even more remarkable than that of the Leblanc rotary pump."

The advantages claimed are as follows:—

1. Extreme simplicity.
2. The small amount of energy required for operating purposes.
3. The high efficiency obtained.
4. Ease with which starting can be effected, and the small amount of attention required whilst at work.
5. Ability to produce the highest possible vacuums.
6. Stability.

* Reprinted from "Recent Developments in Air Pump Design," by E. Jones. Esq., and read before the Institution of Engineers and Shipbuilders in Scotland on 20th March 1917.

Denny-Edgecombe Torsion Meters.

The essential features which distinguish these meters are :—

1. They do not give a "spot" reading, but deal with the whole of the torsional variations during a revolution, and do not make the incorrect assumption that because a steam turbine may be supposed to exert uniform turning moment, therefore the shaft driving a propeller has no variation of torsion. Even with turbine shafting there is often considerable variation under certain propeller conditions.

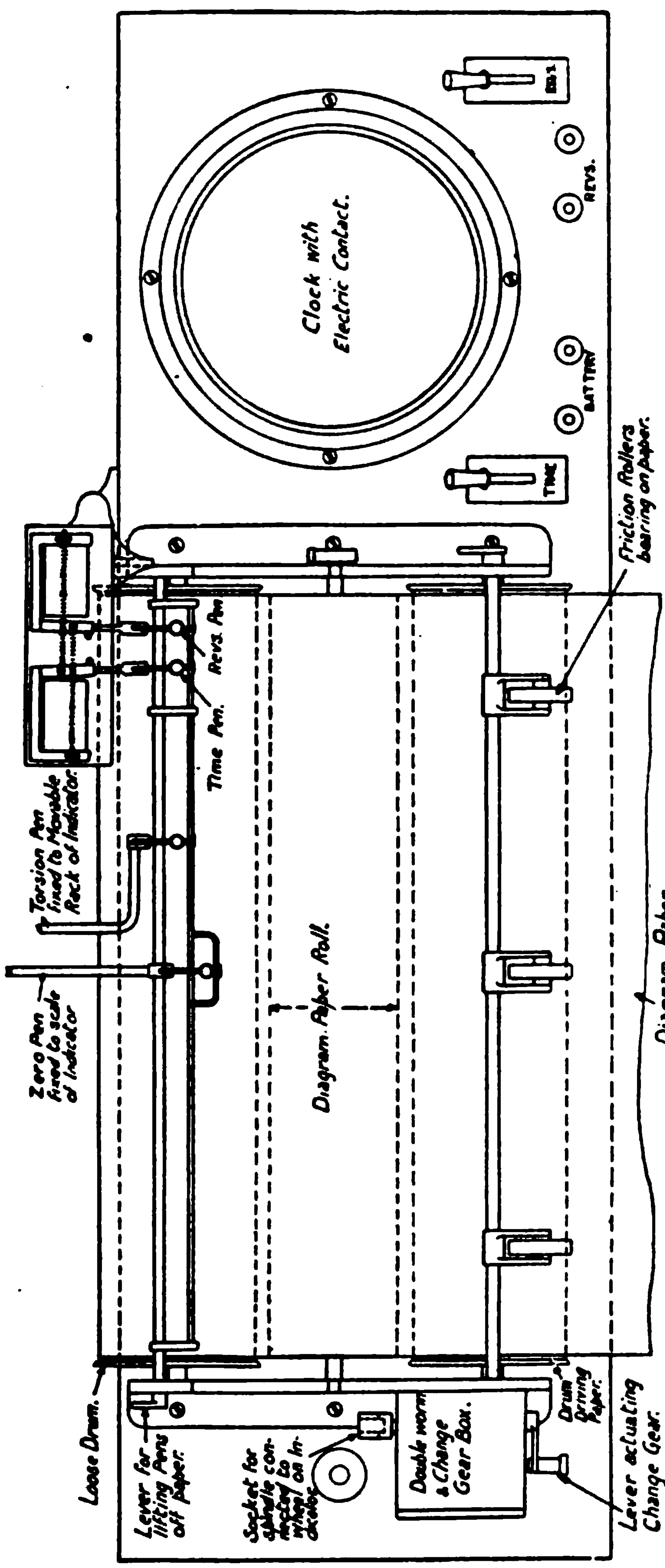
2. They are wholly supported on the shaft, and are not affected by any relative movements of the shaft and its surroundings.

3. The shaft calibration is extremely simple; the meter moves with the shaft, and so long as the moment applied is correctly measured and the shaft is free to twist, the reading remains unaffected by any changes of position other than torsional ones.

General Features in Common.—The meters are of three types, but up to a certain point they are alike. Each consists of a stump and sleeve, made in halves; the stump is secured rigidly to the shaft. The sleeve is only secured to the shaft at one end, *i.e.*, that most remote from the stump. The end adjacent to the stump is fitted with roller bearings, which are adjustable to and roll upon the spigot of the stump. The sleeve at this end has two arms, corresponding to two similar arms on the stump. When the shaft twists, the sleeve remains rigid, and there is a transverse displacement of its arms relatively to those of the stump, corresponding to the twist of the shaft between the two sets of securing bolts. One of the arms has a short rack of involute teeth on its periphery, the other carries an enclosed multiplying gear, the pinion of which can be accurately adjusted to mesh with the rack. This multiplying gear is really a set of toothed levers, which have small angular movements upon their centres. It is convenient to make them in the form of wheels, both for accuracy of manufacture and balance. There are only two spindles; one carries the pinion which gears with the rack, and the other has the final angular movement which it is desired to measure. Both are in ball bearings. At this point the designs diverge.

Type 1 is a purely mechanical meter. The second spindle referred to carries a balanced drum 4 or 4½ in. in diameter. To the drum there is secured, by both ends, a flexible wire rope. The rope passes from the drum, down the arm, round a pulley, along the sleeve, round

No. 2. Indicator.



No. 3.—Recorder.

a pulley on the sleeve, and back again to the drum to which it is securely fastened. The pulley on the sleeve is adjustable to tighten the rope. The circular motion of the second spindle on its axis is thus converted into longitudinal movement of the wire rope.

Traveller.—A light aluminium traveller runs on rollers along a turned part of the sleeve; one pair of its rollers is made to run in a V groove cut in the sleeve. The traveller can thus move along the sleeve, but must revolve with it. It has a turned flange, and a gripper by which it can be rigidly secured to either part of the wire rope in any desired longitudinal position. It is evident that, when the shaft twists, the multiplying gear is moved, and the traveller moves along the sleeve an amount which represents torsion of the shaft. There may also be slight longitudinal movement of the whole shaft, due to expansion or other causes; it is therefore necessary to measure the movement of the traveller relatively to the sleeve. For this purpose there is cast upon the sleeve a

Base Flange, which is turned to the same diameter as the traveller flange. This forms the base of measurement, and also serves as a means of accurately setting the traveller on the sleeve, the two flanges being brought together for this purpose. When the traveller is secured to the wire rope, the difference of position which occurs between the two flanges represents torsion accurately.

The **Indicator** not only measures this, but further multiplies the motion four-fold. It consists of a base plate free to move longitudinally on ball bearings in its case. At the left-hand corner it carries a wheel which runs on the base flange, and is kept in contact with it by a spring so arranged that the indicator base plate follows any longitudinal motion of the base flange, and eliminates everything but torsion. Mounted on the base plate is a small movable carriage which carries at the back a similar wheel, running on the traveller flange; this flange and wheel are kept in contact by another spring arranged on a fusee to give constant tension in all positions. The carriage therefore follows the longitudinal motions of the traveller flange. On the front of the carriage there is a pair of concentric toothed wheels; the smaller of these gears with a fixed rack on the base plate, and the larger gears with a movable rack running in ball bearings also on the base plate. The longitudinal movement of this rack is four times that of the carriage and traveller flange; it has a

pointer which moves over a scale fixed to the base plate, and so registers torsion. In order to increase the fineness of reading, a small

No. 4.—Type 1, with Indicator and Recorder.

spindle is geared to the edge of the larger toothed wheel, and this moves a pointer round a fixed circular disc, one revolution of the

pointer corresponding to two divisions on the fixed scale. If it be desired to obtain a permanent diagram of torsion, a

No. 5.—Type 1a, with Indicator only.

Recorder may be attached to the front of the indicator. This recorder has drums which are driven by means of small double worm

gear from the spindle of the wheel which runs on the base flange. Change gear, actuated by a small lever, is provided, so that the drums can be kept revolving in the same direction whether the ship is going "ahead" or "astern." A diagram paper is thus made to move with the drums, upon which pens attached to the fixed scale and to the movable rack mark respectively a zero line and a torsion line. In addition to this, a clock is fitted with an electrical arrangement for marking half-seconds, and another electric pen is provided which can be connected with a make-and-break fitting on the revolution counter of the engine; a diagram of simultaneous revolutions per minute and torsion is thus obtained.

This type of meter is suitable where the torsion is approximately uniform, and gives a permanent record of changing torsion such as takes place when reversing or changing speed.

There has been a demand for a purely mechanical meter which will follow and record the variations of torsion during a revolution on reciprocating engines. To meet this requirement a modification of Type 1 has been made.

Type 1a is similar to Type 1, in having a wire rope along the sleeve, but in order to obtain the advantages of the purely mechanical type, with the light movement necessary to follow the rapid motions of reciprocating shafts, the traveller, indicator, and recorder of the standard type have been dispensed with in the special meter, the wire is retained, but it is carried round a pulley which is secured to a standard on the base flange of the sleeve, instead of to the sleeve itself. By this device, a non-revolving drum concentric with the sleeve may be slipped along the sleeve until it comes between the standard and the sleeve. The standard carrying the wire thus revolves round the drum. By attaching a light pen rnnning in a slide, to the wire, it will be seen that the pen will trace a line on the drum combined of two separate motions. One is circumferential, and is due to the revolution of the shaft; if there is no load and no torsion, or if there be constant torsion, a straight line, or zero line, will be marked *round the drum*. If, however, the torsion be varying, the pen will traverse its slide in a longitudinal direction, so that supposing the shaft to be twisted but not revolved, a straight line will be marked *along the drum* parallel with the axis of the shaft. When transmitting power, the shaft is both revolving and being twisted, and the pen then traces out a sinuous line, combining the two motions, and thus records the variable line of torsion.

It will be seen that the "pen," as it has hitherto been called, is attached to a revolving part of the apparatus, and is therefore inaccessible when the machinery is running; also, that it moves over the drum at a speed which may be very high on large diameter shafting. The method adopted for dealing with this difficulty is to cover the drum with carbon paper, or other material which will mark a white paper under pressure of a running pen. Over this carbon

paper, on the drum, is secured a sheet of tracing paper. The pen point is a roller with a rounded edge, carried on a spring support. Provision is made by which this roller may be pressed upon the drum, for one or more revolutions as may be desired, and a diagram of the torsional movement results on the back of the tracing paper, but which can be seen through it. There is a second fixed pen in addition to the torsion pen, which gives a base line from which the correct zero line for no torsion may be measured.

It is necessary to know the position of the diagram relatively to the engine; this can be obtained after the torsionmeter is secured on the shaft, by noting the circumferential position of the pen upon the paper when the engine is at some known position, say with the high pressure piston at top of stroke; this point being noted, it can be transferred to the drum, and thence marked on all subsequent diagrams. It may be found convenient to make the pen position coincide with the joint of the paper for a definite position of engine, all other positions of the cycle being thus fixed.

The drum is fitted with rollers which run on the base flange, and is kept in position by a spring catch which adjusts itself to any longitudinal movement of the shaft. The drum therefore follows the shaft, and any error which might be due to longitudinal motion of the shaft is eliminated from the measurement of torsion.

The drum has rollers which run on the turned part of the sleeve, so that when the shaft is revolving the drum is stationary, it is kept from revolving by a casting which slides on a fixed catch bar attached to the ship. There are two lever catches on this bar which hold the drum in one of two positions: (1) against the base flange for obtaining diagram, and (2) away from the revolving standard altogether, so that fresh diagram paper may be put on for the next diagram.

The meter is arranged so that diagrams may be obtained either when going ahead or going astern.

This meter can be used for investigating the variation of torsion on reciprocating engines, either oil or steam.

Type 2 is an electro-mechanical meter, developed from Type 1, in order to enable mean torsion to be read easily in any convenient position, and also to make it "universal." It is used for measuring the power transmitted by any shaft, whether driven by turbines or reciprocating engines. The meter has the same fundamental construction, up to the second spindle of the multiplying gear, as the others.

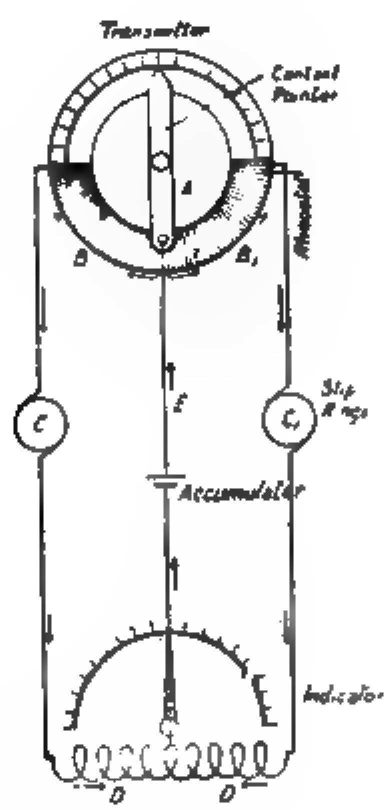
The second spindle in this case carries a light arm, one end of which is a pointer and the other an electrical contacter, and the movement of this arm over a dial fixed to the frame of the multiplying gear is all the work that the gearing has to do. For shafts having uniform torsion there is no motion of the pointer over the dial for constant revolutions, but for reciprocating engines the motion is con-

siderable. The dial has an accurately divided scale for about 180° of its circumference, and the pointer can be adjusted to the zero in the middle of this scale. There is a coiled clock-spring between the spindle and the dial which eliminates the effect of back lash in the teeth of the gear. When the shaft is being twisted in the shop by means of applied moments, the torsion may be read from this scale. When it is revolving in the ship it is impossible to read the scale by eye. Electrical transmission is therefore used to show the reading on a stationary indicator situated in any convenient position.

No. 6.—Type 2.

The transmitter consists of a rheostat imbedded in the dial, with a contacter passing over it and carried on the end of a pointer arm. The rheostat is a thin strip of resistance ribbon of great length ; it is rranged in insulated folds, disposed radially round the dial. The folds of the ribbon are extremely close together, and form a continuous resistance. Electrical connection is made between the two ends of this resistance ribbon and two slip rings on the body of the meter, from which the current is collected by two brushes of special design which ensure uniform transmission of current. Diagram 1 shows diagrammatically the method of transmission. A is the contact finger which is moved round its dial by the multiplying

No. 7.—Type 2, Transmitter showing Rheostat.



No. 8.—Type 2, Diagram showing Transmission.

gear; as the shaft twists, A turns on its centre and the contacter moves over the rheostat BB_1, the ends of the rheostat resistance are connected to two slip rings CC_1, and by means of the brushes the current passes to the ammeter coil DD_1, and back to the accumulator. There are thus two circuits; the current passes from the accumulator by way of the earth wire, which is connected to the bare metal of the ship, and thence by way of the ship and shaft to the contactor A, where it divides and passes the resistances BB_1, to the slips rings CC_1, and on to the two coils DD_1 of the ammeter, coming finally back to

No. 9.—Type 2, Indicator.

the accumulator. It is evident that, as the contacter passes over the rheostat, the resistance in B and B_1, will cause movement of the coil of the differentially wound ammeter, and the pointer of the indicator will follow the contacter of the rheostat.

The **Indicator** is a robust instrument, designed to stand the ordinary vibrations of ships. It has two scales, one being red and the other black; each of these scales corresponds to the one-half of the resistance on the transmitter, the scale of which is also half red and half black, so that distinction may be made for "ahead" and

"astern" readings. These scales on the indicator are constructed by the makers from observed movements of the contacter over the accurately divided scale of the transmitter dial. To assure maintenance of this agreement, a permanent resistance is provided and a "test" switch by means of which the meter circuit can be cut out and a test resistance put in; the position of the pointer on the scale is marked for this condition, and is the standard for "test." The range can be adjusted at any time by the range adjuster even when the ship is running, it is always on the black scale. There are two switches on the bottom of the indicator, one for "ahead and "astern," and the other for "test" and "working." The "ahead" and "astern"

No. 10.—Type 2, Brushes.

is a four-way switch, only two of the positions being in use for any particular shaft, it must obviously be either "red" ahead and "black" astern or *vice versa*. This can be ascertained from the direction of rotation of the shaft; for shafts turning in the usual way, that is outboard on top, the port shaft is always "red ahead" and "black astern," the starboard shaft "black ahead" and "red astern," and this holds whether the meter has its arms either forward or aft. This may be put in another way by saying, that for all shafts driving right handed propellers the ahead readings are on the black scale, while for shaft driving left handed propellers the ahead readings are on the red scale. A small cam is fitted with a screw stopper so that only the requisite pair can be used. The "test" switch has two positions, "working" and "test"; it remains at "working" normally,

but to test the range it is turned to "test," then on pressing the reading key and test key simultaneously the pointer should come to the test mark, using the black scale for full power. When the meter is set up, the only operation necessary to read torsion is to press the "reading" key and take the reading on the red or black scale, as shown by the colour on the ahead or astern switch.

For war vessels, where it is necessary to take very small readings, a "cruising" scale is supplied with an electrical connection to enlarge the lower portion of the scale.

A **Charging Board** is supplied if required, upon which the indicators and accumulators are mounted. There is a charging circuit with lamp resistance to reduce the ship's circuit to charging strength. A spare accumulator is supplied which can be charged and interchanged with either of the others, the indicators being adjusted by means of the "test" switch before described. There is a three-way switch in the circuit, for "Lighting" "Off" or "Charging." In the charging position the lamps will also do for lighting the board.

The current is collected from the slip rings by two special brushes, which need only be put into contact with the rings when readings are being taken. They are mounted on springs in such a way that perfect contact is maintained under heavy vibration. The rings and brushes are occasionally wiped with a rag moistened with paraffin oil.

In all the types, shaft horse-power equals

Reading × Revolutions × Constant.

Method of Obtaining the "Constant."—The usual practice is to twist the shaft in the shop with the meter in position. These meters are wholly supported on the shaft, so that no change of shaft position can affect the result; it is not necessary therefore to maintain the alignment of the centre line of the shaft, neither is it necessary to securely anchor one end of it.

A simple method of calibrating is to hang weights on two levers, one at each end of the shaft, the levers being secured rigidly to the shaft coupling by means of castings. These castings are made circular on the lower side so that they may roll freely on the top of two standards supporting them, the width of this circular part being made adequate for the weight. No shaft bearings are necessary, and no deflection has been noticed with this method. When taking readings the meter should be tapped smartly with a wooden mallet to eliminate any frictional error, special care being taken with the zero reading.

The "constant" is obtained directly from the readings obtained and the turning moments:—

$$\text{S.H.P.} = \text{Reading} \times \text{Revolutions} \times \text{Constant, and}$$

$$\text{also S.H.P.} = \frac{\text{T.M.} \times 2\pi \times \text{Revolutions}}{33000},$$

$$\therefore \text{ Constant} = \frac{\text{T.M. in foot-pounds}}{\text{Reading} \times 5252{\cdot}2},$$

or for metrical units,

$$\text{Constant} = \frac{\text{T.M. in kilogrammetres}}{\text{Reading} \times 716{\cdot}2}.$$

It will be seen that this eliminates the multiplier and the shaft diameter, but gives the correct "constant" from the twist alone.

In Coulomb's formula :—

$$\theta^\circ = \frac{\text{T.M.} \times \text{L}}{\text{K} \times (\text{D}^4 - d^4)},$$

the value of "K" can be got from the readings when twisting by using—

$$\text{K} = \frac{\text{T.M.} \times \text{L} \times \text{Multiplier}}{\text{Reading} \times (\text{D}^4 - d^4)}.$$

Where the T.M. is in foot lbs., L in feet, D and d in inches, the multiplier as supplied with the particular meter, and θ the angular twist of the shaft in degrees.

Method of Obtaining the "Constant" by Calculation.

When it is impossible to twist the shaft, the value of "K" for British units may be taken at about 140, and the constant calculated from the particulars of shaft and meter.

Take Coulomb's formula :— $$\theta^\circ = \frac{\text{T.M.} \times \text{L}}{\text{K} \times (\text{D}^4 - d^4)}.$$

Where θ = Angle of twist of shaft in degrees.
T.M. = Turning moment in foot-pounds.
L = Length of shaft twisted—in feet.
D = Outside diameter of shaft in inches.
d = Inside diameter of shaft in inches.
K = A coefficieut.

or,

$$\theta = \frac{10{\cdot}2 \times \text{T.M.} \times \text{L}}{\text{C} \times (\text{D}^4 - d^4)}.$$

Where θ = Angular displacement in radians.
T.M. = Moment in inch-pounds.
L = Twisted length in inches.
D = Outside diameter in inches.
d = Inside diameter in inches.
C = Modulus of rigidity.

From these formulæ we can deduce the following relation :—

$$\text{Modulus of rigidity} = \text{K} \times 84160.$$

Taking the first formula :—

$$\theta^\circ = \frac{\text{T.M.} \times \text{L}}{\text{K} \times (\text{D}^4 - d^4)}.$$

(*a*) For Type I and Type I*a*—mechanical meters—the reading is linear, for convenience in obtaining diagrams, and the reading at the indicator = the movement at the rack on the arm × multiplier. This multiplier is supplied with the meter.

$$\text{Now } \frac{\text{Circumferential movement at rack}}{2\pi \times \text{Rack radius}} = \frac{\theta^\circ}{360^\circ},$$

$$\therefore \theta^\circ = \frac{\text{Movement at rack} \times 360^\circ}{2\pi \times \text{Rack radius}}$$

$$= \frac{\text{Reading} \times 360^\circ}{\text{Multiplier} \times 2\pi \times \text{Rack radius}}.$$

$$\text{But } \theta^\circ = \frac{\text{T.M.} \times \text{L}}{\text{K} \times (\text{D}^4 - d^4)},$$

$$\therefore \text{T.M.} = \frac{\text{Reading} \times \text{K} \times (\text{D}^4 - d^4) \times 360}{\text{Multiplier} \times 2\pi \times \text{Rack radius} \times \text{L}}$$

$$\text{and S.H.P.} = \text{Reading} \times \text{Revolution} \times \frac{\text{K} \times (\text{D}^4 - d^4) \times 360}{\text{Multiplier} \times \text{Rack radius} \times \text{L} \times 33000}$$

$$= \text{Reading} \times \text{Revolutions} \times \text{Constant}.$$

(*b*) For Type II.—The electro-mechanical meter—the reading is in degrees.

$$\text{Reading}^\circ = \theta^\circ \times \text{Multiplier}.$$

$$\text{T.M.} = \frac{\text{Reading}^\circ \times \text{K} \times (\text{D}^4 - d^4)}{\text{Multiplier} \times \text{L}},$$

$$\text{S.H.P.} = \text{Reading} \times \text{Revolutions} \times \frac{\text{K} \times (\text{D}^4 - d^4) \times 2\pi}{\text{Multiplier} \times \text{L} \times 33000}$$

$$= \text{Reading} \times \text{Revolutions} \times \text{Constant}.$$

SECTION XIV.

CONDENSERS AND AUXILIARIES.

The Weir "Uniflux" Condenser.

A characteristic of the "Uniflux" condenser is its special contour, whereby the entering steam is caused to traverse the cooling surface at practically uniform velocity throughout its passage.

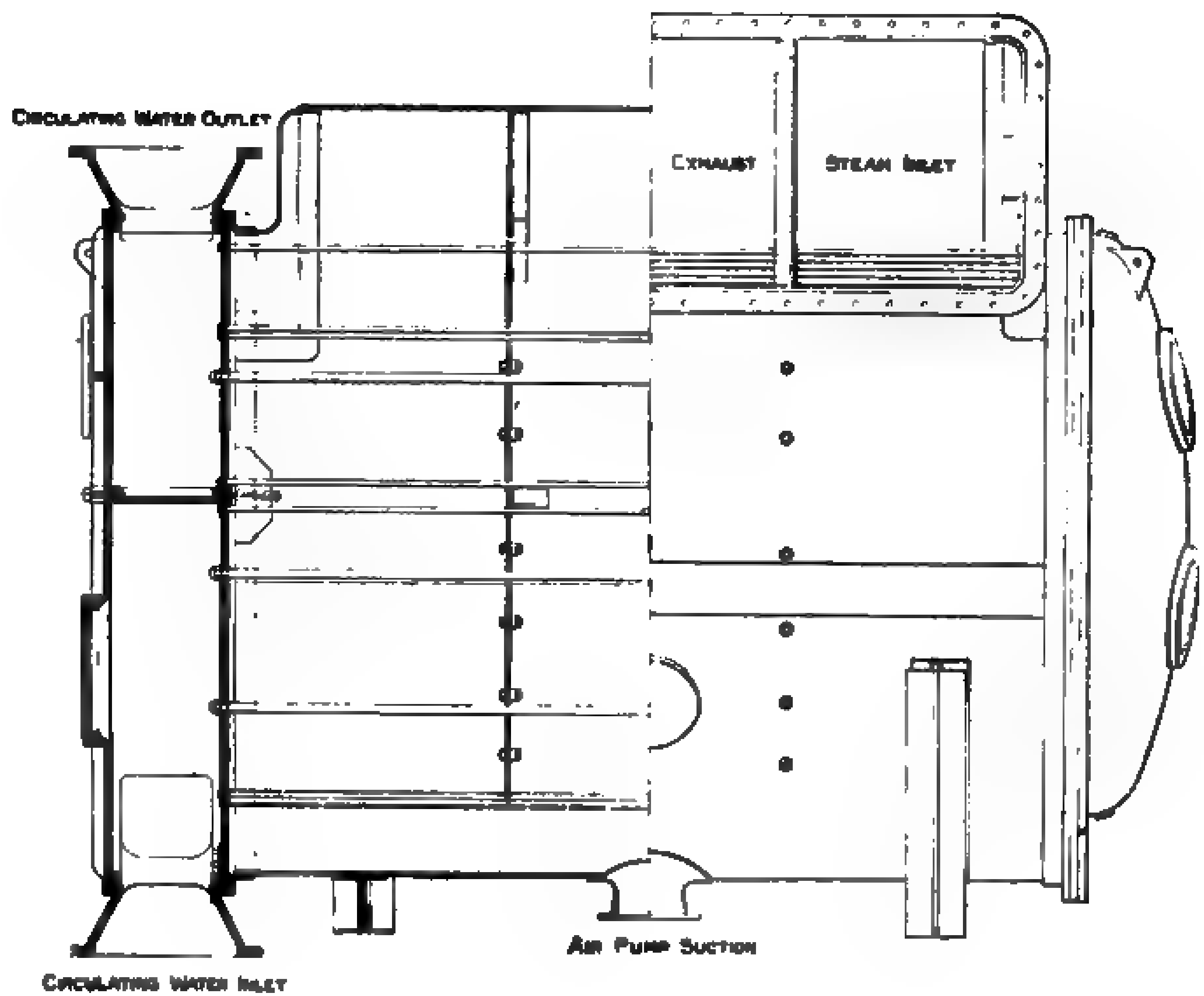
Provision is also made in the bottom of the condenser for ensuring equality of distribution, so that no short circuiting of any portion of the surface can take place. There are no baffle plates or partitions to interfere with the steam flow in the body of the condenser. The flow is therefore direct towards the bottom of the condenser from the exhaust inlet to the draft plate. As a result of the maintenance of the steam velocity the heat transmission value of the condensing surface becomes greatly increased, while by the directness of the flow the whole of the surface is uniformly operated, and no ineffective zones can exist.

As compared to the ordinary type condenser, the Weir "Uniflux" condenser has less cooling surface per H.P., but this is counterbalanced by a slight increase in the quantity of circulating water passing through the tubes, which ensures a more rapid flow. The steam flow through the condenser is also more direct and more convergent from top to bottom than in the usual pattern of condenser as fitted.

The Weir "Dual" Air Pump.

The duty of an air pump is to take from the condenser a mixture of air, water, and vapour; and to obtain economical working, this should be done at the highest possible temperature. If a single pump is used to handle this mixture, the hotwell temperature for a given vacuum is dependent on the amount of air leakage and the air pump capacity. Consequently, with a single air pump the temperature of the mixture must be a considerable degree colder than the theoretical temperature due to the vacuum.

By the use of separate pumps handling the air and water, the dry pump having a cold injection non-returnable to the feed, the above



No. 1.—"Weir-Uniflux" Condenser.

conditions are changed, and higher thermal efficiency is rendered possible.

This method enables the wet (or water) pump to handle water at approximately the steam temperature, while at the same time the dry pump will deal with the air and vapour at the volume and temperature conditions imposed by the temperature of the injection water.

With the "dual" pump the above advantages are obtained

No. 2.—Diagrammatic Section of Weir "Dual" Air Pump, Cooler, and Connections.

together with increased efficiency of the dry pump, due to its contents being densified or decreased in volume through cooling by the injection water, which circulates continuously, never leaves the system, and is never subjected to atmospheric pressure with consequent aeration. The dry air pump further works only at less than half the pressure range, as it discharges below the head valves of the wet pump. The apparatus, moreover, is compact, self-contained and of good mechanical design.

Description.—No. 2 shows in a diagrammatic form the arrangement of surface condenser, "dual" air pump, and injection water

cooler. In all cases the pump A, or wet pump, is situated below the steam cylinder, as this pump is the only one which works under any considerable load; the dry pump B is driven by the beam and links

No. 3.—The Weir "Dual" Air Pump.

in the usual manner. One connection C is made to the condenser, but a branch pipe is led to the dry pump, the connection being made in such a manner that the water will all pass by C′ to the wet pump. Both pumps are generally of the three-valve marine type, but in certain cases the dry pump may be of the suction valveless type.

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1. Steam Slide Valve Chest.
2. Double Joint.
3. Front Stay.
4. Bottom Spindle.
5. Valve Gear Levers.
6. Front Stay Bush.
7. Ball Crosshead.
8. Main Crosshead.
9. Crosshead Pin.
10. Piston Rod.
11. Piston Body.
12. Piston Rings.
13. Cylinder Cover.
14. Discharge Valve Seat.
15. Discharge Valve Seat Ring.
16. Suction Valve Seat.
17. Suction Valve Guard.
18. Discharge Valve Guard.
19. Water Valves.
20. Bucket.
22. Pump Cover.
23. Valve Chest Covers.
24. Steam Stop Valve.
25. Exhaust Stop Valve.

No. 5.—Sectional View, Weir Standard Feed Pump.

The first and most important difference from an ordinary twin pump consists in the dry pump discharging through the return pipe E, through a spring-loaded valve F, into the wet pump at a point below its head valves. The next point concerns the supply of water to the dry pump for water sealing, clearance filling, cooling, and vapour condensing. When starting the pump the filling valve G must be opened for a minute or so to enable the vacuum to draw in a supply from the hotwell of the wet pump. The valve is then closed, and the water passes from the hotwell of the dry pump by the pipe H to the annular cooler, and after being cooled passes into the suction of the dry pump, then, passing through the pump, it becomes heated and again passes to the cooler, and so on in a continuous closed circuit, any excess passing over the pipe E to the wet pump. The spring-loaded valve F is adjusted to maintain about 20 in. vacuum in the dry pump hotwell when the condenser is working at 28 in. vacuum, and this 8 in. difference of pressure is sufficient to cause the water to overcome the cooler friction and pass into the suction, and at the same time never allow any direct air connection between the dry suction and discharge.

Weir's Patent Direct-Acting Feed Pumps.

(To work in connection with Main Feed Pumps and Heater.)

The Weir pump is of the direct-acting type, and has suction and delivery valves for top and bottom independently.

The pump is vertical, single cylinder, and is usually supplied in pairs.

The steam piston is fitted to the top end of the rod, and the water piston to the bottom end, the latter being smaller in area than the former.

Weir's feed pump is a slow-speed, high-pressure, full-stroke pump.

Group Valves.—These are milled out of the solid metal, and are arranged to give a large area with a small lift. Each seat contains a number of small valves, and in all cases these are duplicate, with a lift of ¼ in. The delivery valves have light springs fitted. The suction seat contains a *larger* number of small valves than the delivery seat; it will therefore be noted that the delivery valves have *less* area than the suction valves, and in addition have small springs fitted to keep them down.

The screwed pin in the centre of the valve box covers is for keeping the valve guards in position, and is not for regulating the lift of the valves.

The Weir Steam Valves.—The steam valves of Weir's pumps are simple in action, and the main valve face (known as the "shuttle" valve) is half round instead of flat, and instead of travelling up and down is moved by steam horizontally from side to side. The ports

are therefore arranged to allow of this, and are cast side by side in place of one above the other. The result is, of course, the same, as

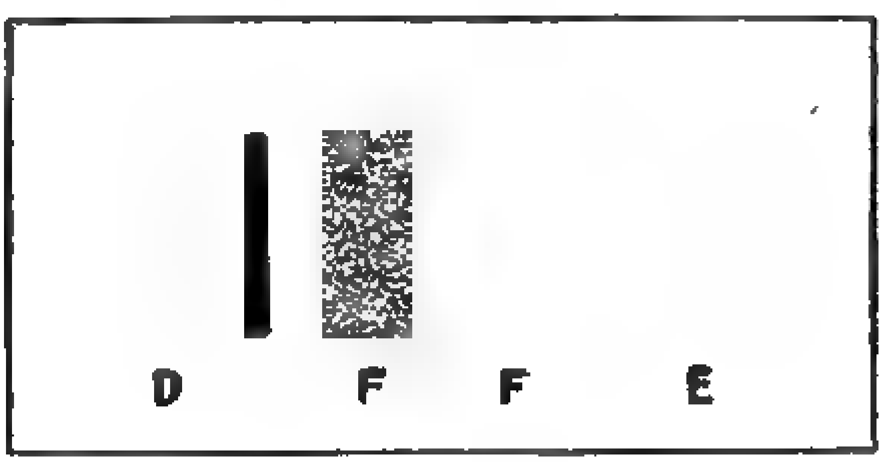
End View. Cylinder Face (Half Round).

A. Steam Port to Cylinder Bottom. C. Steam Port to Cylinder Top.
B. Exhaust Port.

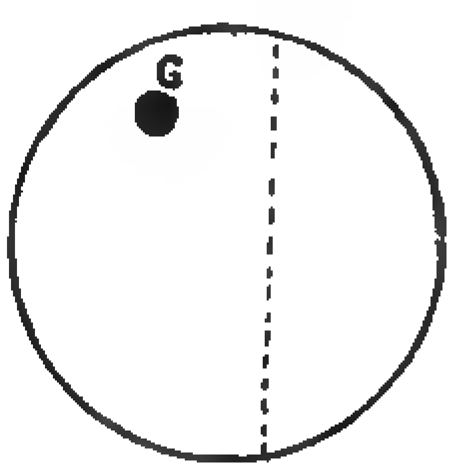
No. 6.—Weir Pump Cylinder Ports.

the left-hand port leads to the bottom of the cylinder, and the right-hand port to the top (see sketch).

Shuttle Valve.—As before stated, this valve is moved by steam hori-



No. 7.—Main (Shuttle) Valve Front Face (Half Round).



No. 8.—End View (Left).

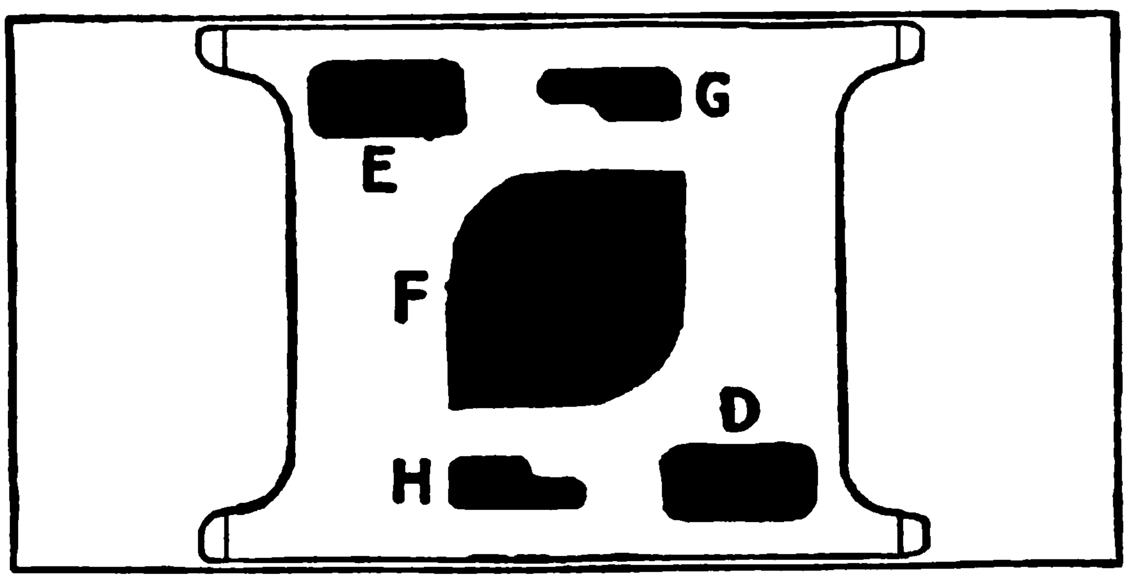
zontally in the chest, and in this way opens up the cylinder ports from steam to exhaust at the end of each stroke. It must be remembered, however, that previous to this the steam passing through the main valve into the cylinder ports has already been cut off by the expansion valve at three-quarters stroke. The ends of the main valve are round, and work in extended cylindrical casings at each side of the chest, the valve being moved across by steam alternately admitted and exhausted from the ends which act as pistons.

The shuttle valve has two faces: that on the front contains four ports, two steam and two exhaust. The face on the back contains five ports (see sketch).

The port E leads from back to front and admits steam to the cylinder bottom by port A.

The port D leads from back to front and admits steam to the cylinder top by port C.

The port G admits steam to, or allows of exhaust from, the *left*



No. 9.—Shuttle Valve Back Face (Flat).

side of the chest in which the round end of the main valve works steamtight.

The port H admits steam to, or allows of exhaust from, the *right*-hand side of the chest in which the round end of the main valve works steamtight.

The centre exhaust port F is common to all the ports. Observe that port G leads to the left-hand end, where a small hole allows the steam to pass out and act on the piston end of the main valve to force it across; it also allows of exhaust to take place from that end to the exhaust port F by means of the small auxiliary or expansion valve.

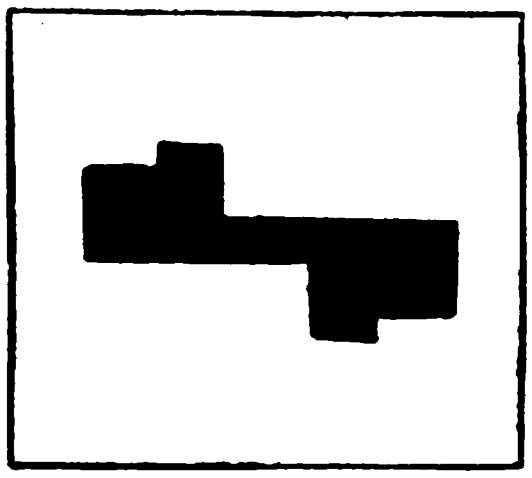
Auxiliary Valve.—This small valve works vertically on the back of the main valve, and admits steam to the ports E D and G H alternately, or allows of exhaust from H and G to port F.

The auxiliary valve is moved up and down by the levers from the pump rod striking a pair of adjustable nuts fitted in the valve spindle, the distance between the nuts allowing of a certain amount of lost

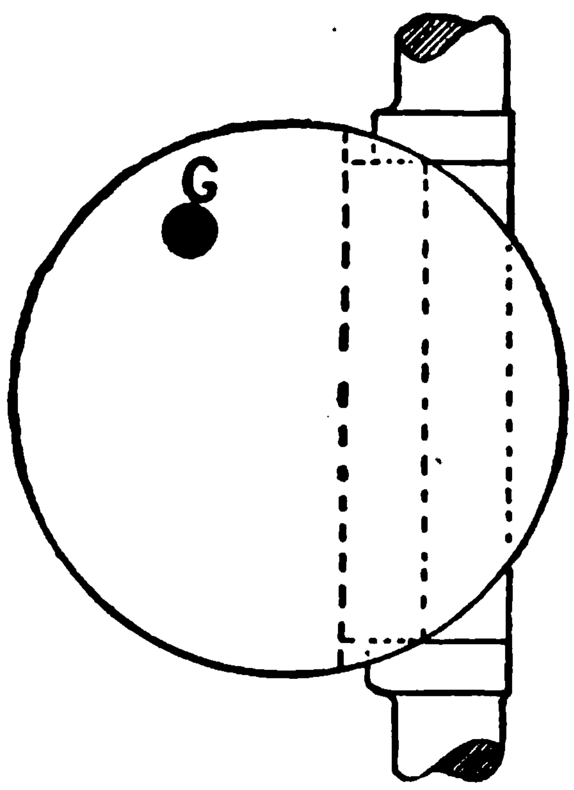
motion. The "lost motion" permits of steam admission for nearly the full length of stroke.

The auxiliary valve has two separate functions :—

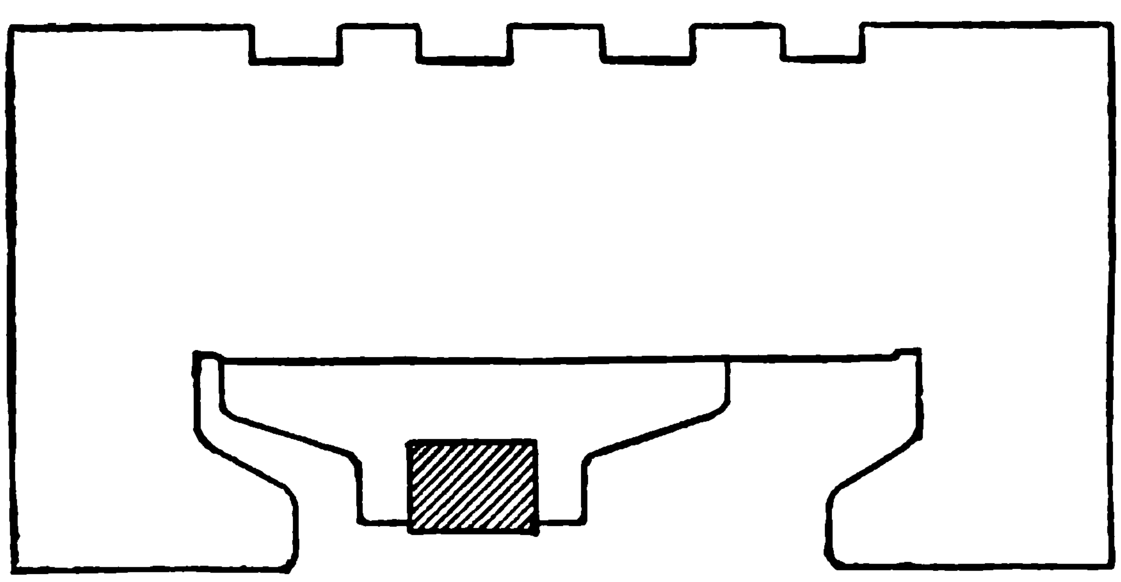
1. To work the main shuttle valve by admitting steam and exhausting it from the valve ends; and
2. To cut off steam at a definite part of the stroke.



No. 10.—Expansion (Auxiliary) Valve Face.



No. 11.—End View of Main and Auxiliary Valves in Position.



No. 12.—Plan of Main and Auxiliary Valves in Position for Steam to Bottom of Cylinder.

Action of Valves (Up Stroke).—When the cylinder piston is on the bottom centre, the main valve is at the right-hand side of the chest, and the auxiliary valve at its lowest position. Steam is then entering through the main valve port E into the bottom port A of the cylinder, and to the left hand end of the main valve by port G: this continues as the piston moves upward until about half stroke, when the lever strikes the nut on the auxiliary valve spindle, and the valve coming up cuts off the steam entering port E at about three-quarters stroke; the steam in the main valve and cylinder then expands and completes the stroke. When the piston reaches the end of the stroke the auxiliary valve opens up the exhaust port G from the left-hand end of the main valve, and the steam acting on the other end, forces the valve across, thus opening the bottom cylinder port A to the exhaust, and at the same time opening the top cylinder port C to steam for the down stroke.

For the down stroke the same process is gone through with the other ports, the main valve moving in the opposite direction, that is, from left to right. It is important to note that the main valve does not move until one end is opened to exhaust: it is then forced across by the steam pressure on the opposite end.

The auxiliary valve after opening the end to exhaust by port G or H, is arranged to close again, so that steam is retained to act as a cushion to the main valve when it flies over, and thus prevent hammering on the chest end covers. It will be seen from the foregoing that the valve gear is positive in action, the main valve being always open to *steam* for one end of the cylinder, and to *exhaust* for the other end, with the piston on the corresponding centre. It is therefore impossible for the pump to stick on the centre points.

Bye-Passes.—Small bye-passes are fitted at each end of the cylinder to admit steam full stroke when required. This may be necessary when starting the pumps, as the cylinder may then contain a quantity of water. "Knocking" can also be reduced by suitable adjustment of the bye-passes.

The bye-passes are formed either by notches cut in the edges of the cylindrical caps, which may be opened or shut by turning round the caps, or by parallel plug cocks, one at each side, which can be adjusted to admit as much steam as required for the occasion.

To Adjust the Length of Stroke of a Weir Feed Pump.

This is done by screwing the valve spindle up into the joint until the piston comes to rest against the cylinder cover, without having raised the auxiliary valve sufficient to throw the main valve. By noting the distance of the crosshead from the gland when the piston is against the cylinder cover, the spindle can be brought down until the valve throws and the piston clearance is ½ in. The lock nut should now be carefully fixed to prevent the spindle from shaking

loose. The bottom stroke adjustment is done by slacking the nuts on the bottom end of the spindle. These should be screwed down until the piston rests against the bottom of the cylinder, and by again noting the distance of the crosshead from the pump gland you can again adjust it for the valve to throw when the pump has $\frac{1}{2}$ in. of clearance. Always run the pump *slowly* when adjusting the stroke. It is most important that any one having the working of a "Weir" pump should become familiar with the method of adjusting the stroke.

General Description of the Weir Turbine-Driven Rotary Feed Pump.

The Weir "Roto-Feed" pump consists of a steam turbine of the impulse type, having one pressure and several velocity stages, working in conjunction with a single stage centrifugal pump of special design.

The turbine casing and bearings are divided horizontally, the lower portions being embodied in the general framework or base of the set. This framework is extended at one end to form a flange, to which the pump casing is securely fastened.

By this arrangement the impeller of the pump is carried on the overhanging end of the turbine shaft; absolute alignment and compactness are thereby secured, and the necessity for a third bearing with its attendant complications is avoided.

It will be clear also that the design readily permits of the opening up of the pump and of the withdrawal of the several parts for internal inspection. The complete turbine casing and base of the set are usually constructed of special close-grained cast iron: the pump is of bronze; and the whole set is carefully designed to ensure lightness and strength.

The upper portion of the turbine casing carries the steam and exhaust branches, and the facing for the nozzle box—this chamber is in direct communication with a series of expanding nozzles by which the steam is guided to the turbine blading. Auxiliary nozzles controlled by valves in the nozzle box are also provided for use in special circumstances.

Admission of steam to the turbine is primarily regulated by a stop valve of ordinary screw-down disc type, provided with dashpot and piston. By a system of levers and trip gear, the piston communicates with a safety or emergency governor of eccentric-ring type directly attached to the turbine shaft, and situated within the shell of the outer bearing.

When the speed of rotation attains the predetermined limit (about 10 to 15 per cent. in excess of the normal) this governor automatically frees the piston; the stop valve is then instantaneously closed by the action of a spring and the steam completely shut off. There is thus no possibility of the speed of the pump reaching unsafe limits in the event of an accidental removal of the load.

Provision is also made for regulation of the steam supply to the turbine under the conditions of fluctuating load which normally prevail in power installations. This is carried out by means of a ratio governor of the differential-piston type. This governor operates a double-beat balanced throttle valve adjoining the main stop valve, and interposed between the latter and the nozzle box of the turbine.

This governor is subjected, by means of the differential piston, to the simultaneous action of the steam pressure at the turbine and the discharge pressure of the pump, and acting on the throttle valve it

No. 13.—Weir Rotary Feed Pump.

regulates the steam opening and speed of the pump so as to maintain a constant ratio between these pressures, whatever may be the actual output of the pump.

The pump thus automatically adjusts itself to changes in both water and steam conditions. When the latter are decidedly adverse, or when the pump is so heavily loaded that the ratio governor is in its uppermost position and the throttle valve full open, the auxiliary nozzles may be called into play by means of the auxiliary valves on the nozzle box. These valves, which are placed vertically in order to facilitate the easy inspection of the outer bearing of the turbine,

may be employed to augment considerably the quantity of steam supplied.

The turbine glands are provided with carbon packing, which can be readily removed without dismantling the shaft; the turbine bearings are of the self-oiling ring type, and one of them is designed as a guide bearing. It operates in this way, however, only in starting and stopping the set, as during running the end thrust is completely balanced by a pressure-equalising device within the pump itself.

Subject to the temperature conditions, the pump is capable of lifting water from the maximum possible depth, and may be arranged to discharge against the highest boiler pressures.

The casing of the pump is of volute form, the suction being taken from the side; the impeller is of the single-suction shrouded type. The water flows into the pump in an axial direction, and is delivered radially in the customary manner. By suitable adjustment of the casing relative to the bolt holes in the main frame, the discharge branch may be set in any one of a series of equiangular radial positions according to the number of bolts employed.

In order to secure absolute accuracy of form and workmanship, and true alignment, the packing rings for the pump impeller are so constructed that they may be moved axially along the impeller, and the accuracy of their construction and fit ensures the highest efficiency.

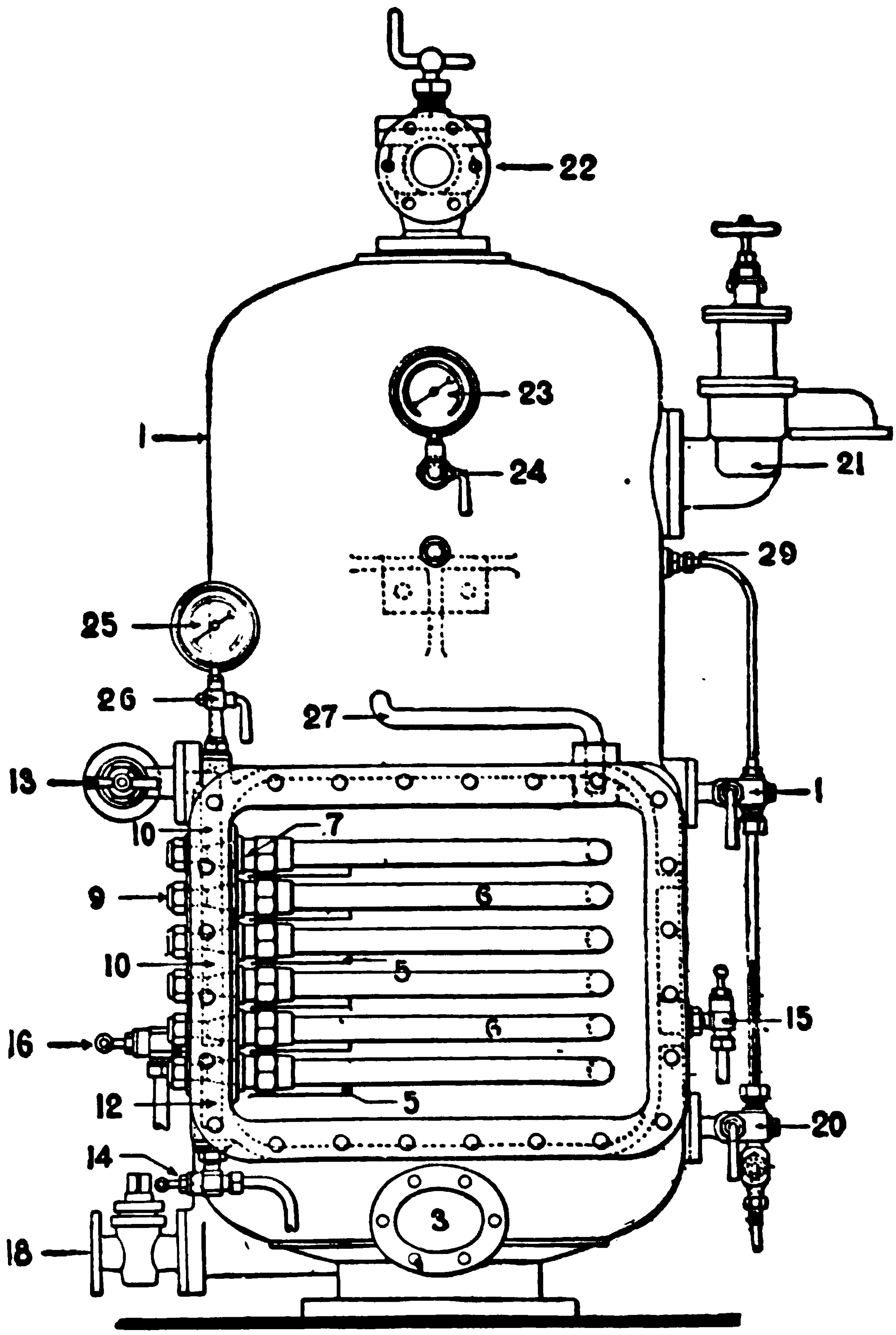
The turbine casing is provided with a compound pressure gauge, and a sentinel valve normally set to 30 lbs. per sq. in.—these fittings serve to indicate any irregularity in the working of the turbine. Arrangements are made for temperature readings by the insertion of thermometer plugs on the nozzle box and on the turbine bearings.

"Weir" Evaporator (14, 15).

The Weir vertical type evaporator is a single casting of close-grained cast iron. The heating surface is formed of heavy solid drawn copper tubes attached by hollow conical couplings, at one end to the steam inlet chamber cast on the side of the evaporator, and at the other end to the corresponding outlet chamber cast alongside it.

The tube space is separated from the steam space by a deflector. The arrangement of this deflector allows the steam to rise, but throws down the water so that it is returned again to the water space. With this arrangement it is almost impossible to make the evaporator prime under anything like reasonable limits.

The usual mountings consist of steam inlet valve, steam outlet valve, feed check valve, brine valve, drain valve and coupling to hot-well, blow-off cock, safety valve, gauge glass fittings, pressure gauge, compound gauge, salinometer valve, also gun-metal feed pump to work off main engines.



No. 14.—Weir Vertical Type Evaporator.

No 15—Weir Vertical Type Evaporator.

List of Parts of Evaporator.

1. Shell of evaporator.
2. Main door of evaporator (1) for withdrawing tube coils (6).
3. Hand cleaning door.
4. Baffle plate, or deflector.
5. Shelves for supporting tube coils (6).
6. Evaporating tube coils.
7. Inlet steam couplings for coils (6).
8. Drain outlet couplings for coils (6).
9. Coupling nuts for 7 and 8.
10. Inlet steam header.
11. Drain header.
12. Drain collecting pocket.
13. Inlet valve for steam coils (6).
14. Valve for drain from coils (6) to hot-well.
15. Feed check valve.
16. Brine valve.
17. Salinometer cock.
18. Cock for blowing off to sea.
19. Top cock for water gauge.
20. Bottom cock for water gauge.
21. Safety valve.
22. Outlet valve for generated steam.
23. Compound gauge for generated steam in shell (1).
24. Cock for compound gauge (23).
25. Pressure gauge for inlet steam to coils (6).
26. Cock for pressure gauge (25).
27. Swing crane bar for door (2).
28. Eye bolt for supporting door (2) on crane bar (27).
29. Connection from top of water gauge (19) to steam space in evaporator shell (1).

Weir Surface Type Feed Heater (16).

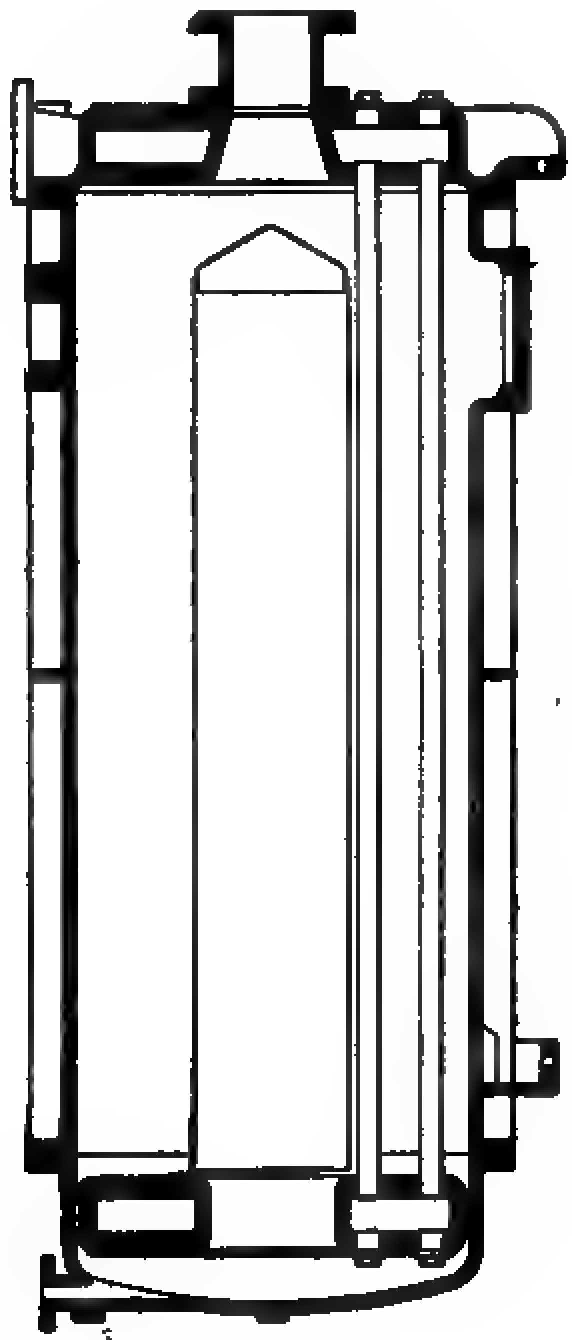
To reduce the risk of oil or grease from the heating steam finding its way into the feed water and setting up priming in the boilers, a surface type of feed heater, in place of a direct contact feed heater, is sometimes preferred for turbine work. Impurities passing over with the boiler steam may cause serious troubles in the turbine casings, such as corrosion and blade stripping, etc.

The Weir surface heater is of the tubular type, the heating surface consisting of copper tubes of ample diameter expanded into headers top and bottom, the feed water circulation outside the tubes being specially arranged to give the maximum heat transmission possible. Water bye-pass valves are fitted to allow of direct feed to the boilers when overhauling the heater.

The direct contact feed heater is more often fitted in turbine steamers of the merchant service, and it may be mentioned that this type is rather more efficient than the surface heater, as the heat transmission losses through the tubes are eliminated. The upkeep and attention required is also less owing to the fact that tubes are done away with in this type.

Weir Contact Type Feed Heater (17).

In this well-known type of feed-water heater, the heating steam is taken from the low-pressure receiver of the main engines and exhaust of the auxiliary engines, and enters the heater through a non-return valve on the side; the feed water is forced up into the top of the heater by the main feed pumps on the engine, and, the pressure of the water overcoming the tension of the spring, forces open the internal valve and allows the water to enter the body of the chamber, through a perforated ring in the form of a fine spray, which, meeting the heating steam entering by another set of perforations,



No. 16.—"Weir Multiflow" Surface Type Feed-Water Heater.

becomes raised in temperature to that of the steam: the air present in the water is set free by the heat, and rising, escapes from the heater by a cock on the top to either the condenser or the atmosphere. The float shown near the bottom of the heater is connected by a system of

No. 17.—"Weir" Direct Contact Feed Heater.

levers to the steam stop valve of "Weir's" pumps, which take away the hot feed water and deliver it into the boilers, and as the water level in the chamber falls, the float sinking shuts off the steam supply to the pumps, so that they work slower, and, on the other hand, when the water level is high, the float rising correspondingly opens the steam valve, and the pumps work faster in proportion.

Two pressure gauges are fitted on the top of the heater, one to indicate the water pressure entering, and the other to indicate the pressure in the body of the chamber, upon which latter depends the temperature of the hot feed water.

NOTE.—With a pressure of 5 lbs. in the L.P. receiver, the feed temperature in the heater will be about 220°.

Advantages of Feed Heating.

The practical advantages of feed heating are as follows:—

1. Less straining of the plates in high-pressure boilers.
2. Less air enters the boilers, and consequently the corrosion due to oxygen is reduced.
3. The boilers steam better with hot feed water, as circulation is checked when cold water enters a boiler.
4. The heat of condensation of the steam is given up to the feed water, instead of being given to the circulating water and lost over the side, as would be the case if the steam condensed in the condenser.

Feed-Water Filters.

A feed-water filter is employed to prevent oil and grease from entering the boilers with the feed water. It is usually placed between

No. 18.—Feed-Water Filter.

(American Steam Gauge and Valve Manufacturing Co.)

the feed pumps and the feed check valve, and consists of a cast-iron chamber containing a series of perforated brass plates with filter cloths fitted between them, or brass grids arranged in a similar manner.

The feed water is forced through the perforations and cloths, and leaves behind the greasy matter, which is blown out at intervals.

The fittings on the filter are:—Bye-pass valve (used when cleaning out or changing the cloths), safety valve, pressure gauge, soda cock, and drain cock.

Sometimes, instead of cloths, ashes or charcoal are used as the filtering medium.

Salinometer and Hydrometer.

COMPARISON OF DENSITY MEASURING INSTRUMENTS.

INSTRUMENT.	DIVISIONS ON STEM.	
(1) Ordinary Salinometer. (Temp. for use = 200° Fahr.)	Each division = 1 oz. per gall., or $\frac{1}{5}$ density of sea water. 1 degree = 5 oz. = density of sea water between $\frac{1}{32}$, $\frac{1}{33}$.	Limit of indication usually 5 degrees, or $\frac{5}{32}$ density.
(2) Hydrometer. (Temp. for use = 200° Fahr., and Temp. for use = 100° Fahr.) Is usually graduated on one side for 100° Fahr., and on the other for 200° Fahr.	Each degree (division) = ·5 oz. per gall., or $\frac{1}{10}$ density of sea water. 2 degrees = 1 oz. per gall., or $\frac{1}{5}$ density of sea water. 10 degrees = 5 ,, ,, = density of sea water (between $\frac{1}{32}$, and $\frac{1}{33}$).	Limit of indication usually 40 degrees, or $\frac{4}{32}$ density.
(3) Sensitive Hydrometer. (Temp. for use = 100° Fahr.)	Each division = ·05 oz. per gall., or $\frac{1}{100}$ density of sea water. 2 ,, = ·1 ,, ,, $\frac{1}{50}$,, ,, 10 ,, = ·5 ,, ,, $\frac{1}{10}$,, ,, 20 ,, = 1 ,, ,, $\frac{1}{5}$,, ,, 50 ,, = 2·5 ,, ,, $\frac{1}{2}$,, ,, 100 divisions (one degree) = 5 oz. = density of sea water (between $\frac{1}{32}$ and $\frac{1}{33}$).	Limit of indication usually 2 degrees, or $\frac{2}{32}$ density.

NOTE.—To obtain accurate density readings it is necessary that a thermometer should be used conjointly with any of the above-mentioned instruments to ensure that the temperature of the water corresponds exactly with that marked on the stem, as, should the temperature fall below this, the reading will then be in excess of the true density to the extent of approximately $\frac{3}{4}$ ounce for each 10° drop. This correction allowance would therefore require to be afterwards deducted to obtain the actual density.

EXAMPLE.—Suppose the weight of a gallon of boiler feed water to be 170 ounces (that is, 10 ounces per gallon density), and when testing the water with the hydrometer the temperature fell from 200° (the correct temperature) to, say, 190°. Find the density reading obtained, also the true density.

Then (from table of water volumes) we find that—

$$\text{Water at } 32^\circ = 1 \begin{cases} 200^\circ \text{ Fahr. the relative volume} = 1{\cdot}0389. \\ 190^\circ \text{ Fahr.} \quad ,, \quad ,, \quad = 1{\cdot}0350. \end{cases}$$

So that, $\frac{1{\cdot}0389 \times 170}{1{\cdot}035} = 170{\cdot}73$ = reading obtained. As temperature has dropped 10° (200° to 190°), then allow a correction deduction of ¾ ounce.

Therefore, $170{\cdot}73 - {\cdot}75 = 170$ (approx.) ounces density.

Phenol-phthalein Test for Alkalinity.

A few drops of this liquid added to a sample of water weighing, say, about ½ lb., will indicate an alkaline condition by changing the sample to a bluish-pink colour; if, however, no change in colour takes place, then the sample is acid in condition.

For boiler feed water the alkaline condition is desirable.

Efficiency of High-Pressure Steam and Low-Pressure Steam Compared.

$$\text{Heat Efficiency} = \frac{T^\circ - t^\circ}{T + 461} \text{ (see p. 398).}$$

Where, T° = temperature of steam.

t° = ,, ,, exhaust.

461 = absolute temperature constant.

Take steam of, say, 80 lbs. gauge pressure, or 95 lbs. absolute; also steam of, say, 180 lbs. gauge pressure, or 195 lbs. absolute, with exhaust steam temperature of 160° in each case.

Then, 95 lbs. = 324° Fahr. (from table).

And, 195 ,, = 380° ,, (,, ,,).

Exhaust temperature = 160°.

Low Pressure—

$$\text{Efficiency} = \frac{324^\circ - 160^\circ}{324 + 461} = {\cdot}209 \text{ (nearly).}$$

High Pressure—

$$\text{Efficiency} = \frac{380^\circ - 160^\circ}{380^\circ + 461} = {\cdot}261.$$

$$\text{Per cent. economy} = \frac{({\cdot}261 - {\cdot}209) \times 100}{{\cdot}209} = 24{\cdot}8 \text{ per cent.}$$

So that the theoretical economy of the higher pressure is 24·8 per cent. over that of the lower pressure. In actual practice the gain is about 15 per cent.

Coal Consumption, I.H.P., Speed, and Distance Run.

In comparing coal consumption, speed, and distance run, it should be remembered that within moderate speeds the coal consumption

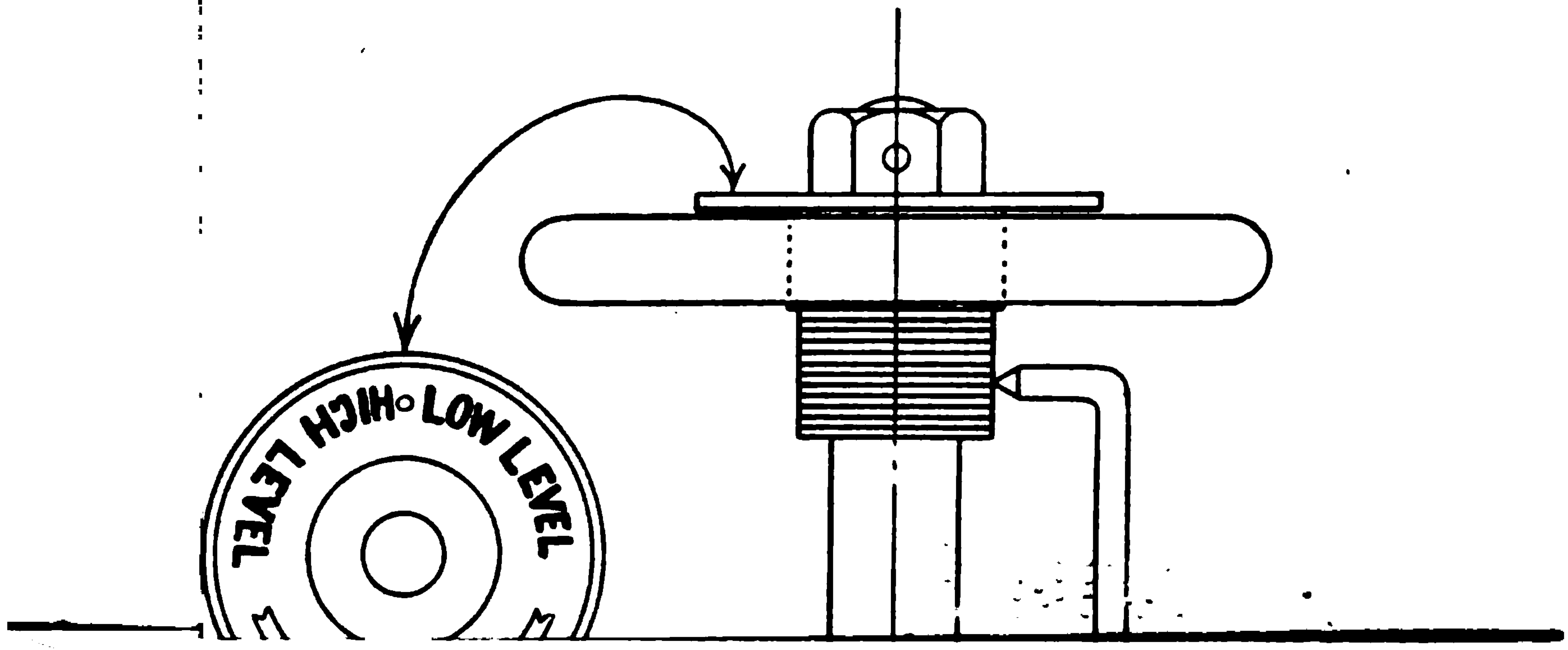
(or I.H.P.) varies as the speed^2 × distance, which means that the coal burnt per knot varies as the speed.^{2} This amounts to the same thing as stating that the consumption varies as the cube of the speed. Therefore we have the following laws which apply to moderate speeds for any given steamer:—

1. The consumption (coal or water) (or I.H.P.) varies as the speed.^{2}

2. The consumption per knot varies as the speed.^{3}

3. The consumption over a voyage varies as the speed^2 × distance.

NOTE.—**Above a certain speed limit the I.H.P. may vary as the 4th, 5th, or 6th power of the speed. From the foregoing it will be seen that if a steamer runs short of coal, port may be reached if the speed is reduced, for although the time taken is much longer, this is more than balanced by the reduced daily consumption, which, under the reduced speed conditions, may be found sufficient to last the voyage.**



boilier and float chamber.

for steam and water connections to

dle valve chest back to feed tank,
en the water is at normal working

ng or testing.

which can be varied by the hand
levels.

, and check piston **B**.

ton **B** for water.

regulating needle valve for different

an easy fit in the cylinder **C**, so
and flows through pipe **E** into
opening and hollow spindle **Q**,
back through pipe **M** to the
ction. This action takes place
, and in rising pulls down the
ts in communication the top
bottom outlet port by means

[*To face page* 630.

SECTION XV.

OIL FUEL.

This method of firing has recently come very much to the tront, particularly in naval practice, and a brief description of the system will not be out of place. The chief drawbacks to the use of oil fuel at present are those of supply and of cost; oil supply ports being few in number, although in due time this matter will be remedied. The cost is also a consideration, but this will also be adjusted to meet the requirements of the demand. It may safely be stated that as fuel, oil has a great future before it in marine practice.

Advantages of Oil Fuel over Coal.

1. Less bunker space required (about 36 cub. ft. against 44 cub. ft. per ton).
2. Greater heat per pound (20000 B.T.U. for oil against 14500 B.T.U. for coal).
3. Cleanliness both in working and bunkering.
4. Reduced stoke-hole staff.
5. Greater control of fires.
6. More complete combustion obtained.

Disadvantages.

1. Difficulty of obtaining oil supplies.
2. Cost.
3. Danger from inflammable vapour caused by leakage into bilges, etc.
4. Danger of oil leaking into steam side of heater and finally entering boilers.

Oil and Coal Compared.

Fuel.	Heat units per pound.	Bunker space per ton.
Coal - - -	14500	44 cub. ft.
Oil - - -	19000	36 „ „

NOTE.—Taking into account both heating value and bunker space, one ton of oil is equivalent to 1·6 tons of coal.

The U.S. Naval Department Committee report on the advantages of oil fuel as follows:—

1. A greater evaporative efficiency in ratio of about 14 to 9.
2. A reduction in the fire-room force.
3. Elimination of ashes and dirty fires.
4. Convenience of storage. Oil tanks may be located in double bottoms, or in spaces now useless.
5. Rapidity and ease of taking aboard and handling. The manual labour in this connection is eliminated.
6. Easy control of fires, permitting sudden variations in power developed by boilers.
7. Facility of controlling proportions of the air and fuel, thus ensuring good combustion. There is no opening or shutting of furnace doors of varying thicknesses as is the case with coal.
8. Elimination of cinders and of smoke, except at full power.
9. The reduction of fire room, there being no space required to permit working of the fires.
10. As there is still a much better distribution of coal among the seaports of the world than oil, this is said to be one of the principal disadvantages of oil.

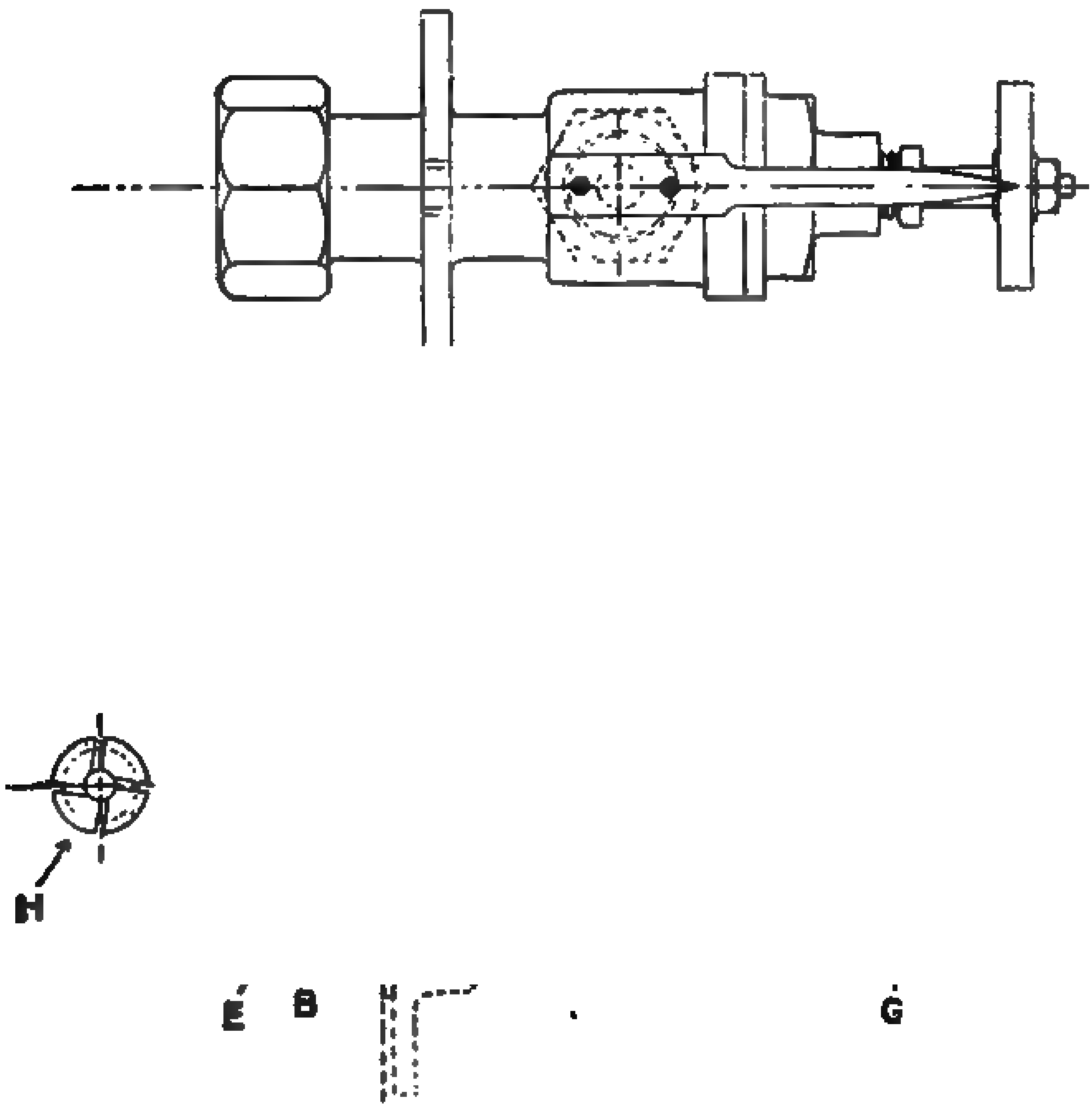
Composition of Oil.—The average composition, etc., of the oils used as fuel are as follows:—

Class.	Flash Point.	Specific Gravity.	Carbon (C).	Hydrogen (H).	Oxygen (O).	Heat Units per Lb.
			Per Cent.	Per Cent.		
Burmah	200°	.92	86	12	1.5	18800
Shale	125°	.81	86	13	1	19000
Russian Petroleum	120°	.822	86	14	0	20000

Methods of Working.—Oil is pumped into the tanks which act as bunkers, and is afterwards pumped into direct supply tanks known as "settling tanks." These tanks are generally placed at some height above the floor plates, and are intended to allow any water which may have become mixed with the oil to settle to the bottom, leaving the pure oil on top. This oil, after being heated and filtered, is then pumped under pressure direct to the burners which are fitted on the front of the furnaces.

Oil Spray.—The oil, under pressure, is sprayed into the furnace through the nozzle and needle valve of the former, and this results in atomising of the oil, which flashes up just after leaving the nozzle point, the effect produced being that of a mass of gas at white heat filling up the entire space of the furnace or combustion chamber.

Burners.—In one system the oil jet is mixed with steam as it enters the burner, and in another system air only is mixed with the oil jet and forced through the burner.



No. 1.—"Kermode" Type Oil Burner.

A, Oil feed to burner.	E, Cap nut.
B, Nozzle.	F, Graduate wheel.
C, Regulating spindle.	G, Pointer.
D, Burner body.	H, Grooves in nozzle end.

Shale Oil.—With Scotch shale oil a heating temperature of about 125° Fahr. is generally sufficient.

Specific gravity of CO = ·96.

Working Oil Fuel.—The oil fuel is pumped from the supply tanks by the oil pumps, and forced through a cold filter, then through a steam heater where the temperature is raised, next through a second or hot filter, and from there to the distribution header on the boiler front, from which valves and pipes connect to the several burners fitted. In Yarrow type boilers from eight to eleven burners are supplied, but of course all of these may not be required except when steaming under full power conditions. The pressure of the oil is about 100 lbs., and the temperature 200°, with "shale" or "Texas" oil as fuel. ***For the successful working of oil fuel, the temperature of the oil is of vital importance, and must be maintained at all costs.***

Control.—The regulation of the fires is controlled by the needle valve of the burner, which can be altered to increase or decrease the angle of discharge, and therefore the output or consumption; one small wheel on the burner constituting the entire control gear. The combustion of the oil is therefore regulated by the oil needle valve and the air supply doors.

The oil fuel is pulverised by being forced, when heated up, under pressure, through the restricted opening of the burner end, which, by means of the grooves, imparts a rotary motion to the jet, the latter being distributed in a cone-like spray or cloud of pulverised oil particles.

Starting Up.—In starting up the fires, a hand pressure pump is employed, which forces a small quantity of oil through a **U**-shaped and flexible tube which is placed inside the air cone; and, having previously set fire to a quantity of oil waste, the heat so produced raises the temperature of the oil flowing through the **U** tube on its way to the starting burner. After the starting burner has been the means of raising sufficient steam to set away the oil pumps, the starting device described is withdrawn, and the other burners are set away. About three hours is allowed to get up steam in water tube boilers.

Leakage Test.—Before the fires are started a leakage test of the oil system is generally made by raising a pressure of 50 lbs. or more with the hand pump, and examining for leaks at the tanks, pipes, joints, bilges, etc.

Colour of Gases.—The colour of the gases in the combustion chamber space indicates the efficiency of the combustion taking place inside, and the pressure and temperature of the oil, in addition to the output required, is regulated accordingly. A very small tail of smoke at the funnel top indicates that combustion is practically complete. The colour of the gases can be observed by means of sighting holes in the boiler casings.

Flash Point and Firing Point.—It should be noticed that the spray of atomised oil at a flash point of 200° is below the "firing point," but on striking the cone ring, the temperature of which exceeds the flash point, ignition instantaneously takes place.

Firing Point.—The firing point of oil is above the flash point, and means that the oil itself (instead of the vapour) ignites.

Flash Point.—By this is meant the temperature at which the vapour formed from heated oil flashes into flame when brought in contact with a light. The flash point varies from 70° in light petrol spirit to about 240° in heavy burning oils. The Board of Trade require a flash point of not less than 185° Fahr.

Sand.—As a safeguard against fire, boxes of sand are placed ready for use in the stoke-holes, as the sand thrown on oil flame quickly extinguishes the same.

Black Smoke.—This is caused by the temperature or pressure of the oil fuel being too low for complete combustion.

White Smoke.—This may be caused by excessive air supply or by faulty oil feed through the burner.

Ventilation Pipes.—To allow for the escape of oil vapour "swan-neck" pipes should be led from the top of the oil tanks to the deck, the ends of the pipes being open, but covered with gauze wire to reduce risk of explosion by a naked light. When the tanks are empty, however, the vapour formed by slow evaporation is much heavier than the atmosphere, and will therefore occupy the *lowest* position in the tanks, and the gases thus formed are best removed by means of exhausting fans.

Air Vessel.—To maintain a steady oil pressure in the sprayers an air vessel is fitted on the discharge side of the oil supply pump.

Settling Tanks.—Sometimes these tanks are fitted with a steam coil to heat up the oil, the effect of which is to separate more quickly the water; the heating up causing a greater difference in the respective densities of the two liquids. The heating is done by exhaust steam of low pressure and temperature, so that there will be no danger of the oil vaporising. A gauge glass is fitted on the tanks, and the water shown can be drained off by suitable drain pipes.

A temperature of 180° is required to produce separation of the water from the oil in the settling tanks.

Evaporation of Oil.—1 lb. of oil fuel evaporates about 15 lbs. of water into steam (from and at a temperature of 212°).

Water in Oil.—In burning oil fuel, water shows by the oil forming a brown coloured foam near the burner nozzle. Sputtering also occurs at the burner, and if the water present is excessive, the burner flame may go out altogether.

White Vapour.—In burning oil fuel, white vapour at the funnel top indicates that the oil vapour is passing off unconsumed owing to excessive air supply, which lowers the temperature of combustion, with the result stated.

SECTION XVI.

BOILERS.

Babcock Boiler (1-2).

The Babcock & Wilcox boiler is of different construction, the tubes being expanded into boxes usually termed headers. These headers are fitted with small doors in line with the tubes to allow of cleaning same. The headers or sections are again connected to the top or steam drum. This boiler has also a large grate surface, and is best fired on the same principle as the Yarrow, that is, a level fire of 6 or 7 inches thick. This boiler has the advantage of working on natural draught, and is a good steaming boiler under those conditions. As in the Yarrow boiler, the circulation takes place in a similar manner, downcast tubes being fitted leading back to the mud drum or bottom header. Blow-off cocks are fitted on this header for cleaning out header. The furnace in water tube boilers is lined with fire-brick throughout, and the ashpits are kept supplied with water so as to avoid damage to the fire-bars. Superheaters are being fitted in the Babcock & Wilcox boiler, superheating the steam 100° to 150° above the saturation temperature. The superheater is fitted athwart-ships, and consists of a series of U-shaped tubes connected to headers in a similar manner to the main boiler tubes. After steam is generated in the main boiler it is passed through the superheater before passing to the machinery. Steam connections for cleaning tubes of soot are fitted, access being obtained by doors on each end of boiler. Zinc plates in perforated holders are fitted in the steam and water drum, and also in the bottom or mud drum. These boilers are possessed of several advantages, steam being quickly raised, but to maintain this efficiency it is necessary to keep the boiler in a clean condition, which necessitates the cleaning of the outside of the tubes as often as required. This operation can be carried out while boiler is steaming, as in the Yarrow boiler.

Water tube boilers require to be kept as clean as possible owing to the possibility of the tubes silting up, and require constant attention as regards treatment with lime, etc.

The two lower rows of tubes are about 4 in. outer diameter, and the others 2¼ in. outer diameter; the larger size of the lower rows reducing the tendency to buckling caused by the intense heat at this position. The severe expansion of the tubes nearest to

the flame acts to produce the bending referred to, and is a source of trouble in nearly all water tube boilers, although in certain types suitable allowance is made to take up the unequal expansion effect.

The larger size of the lower tubes also minimises the danger of

LONGITUDINAL SECTION

No. 1.

choking up of the tubes through scale deposit, which, if the feed contains sea water, quickly forms on the heating surfaces. Both the Yarrow and the Babcock boiler are specially adapted for the burning of oil fuel, which is now the usual practice in the Navy.

A general idea of the construction of the "Babcock & Wilcox"

water tube marine boiler will be obtained by referring to the illustrations Nos. 1 and 2, which clearly show the various parts.

The boiler is constructed entirely of wrought steel, and consists of a series of straight water tubes placed in an inclined position, under

FRONT ELEVATION SECTION AT A B

No. 2.

which the furnace is situated; the tubes are expanded at each end into boxes of sinuous shape called "headers." Opposite each tube there is a separate handhole in the "headers" for inspection and cleaning of the tube, and it is peculiar to note that no stay tubes are fitted. The upward and downward headers are in communication at the top end with the steam and water drum, the downward headers

being connected with a mud drum at the bottom end, which is fitted with blow-off cocks for clearing out the sediment which collects there.

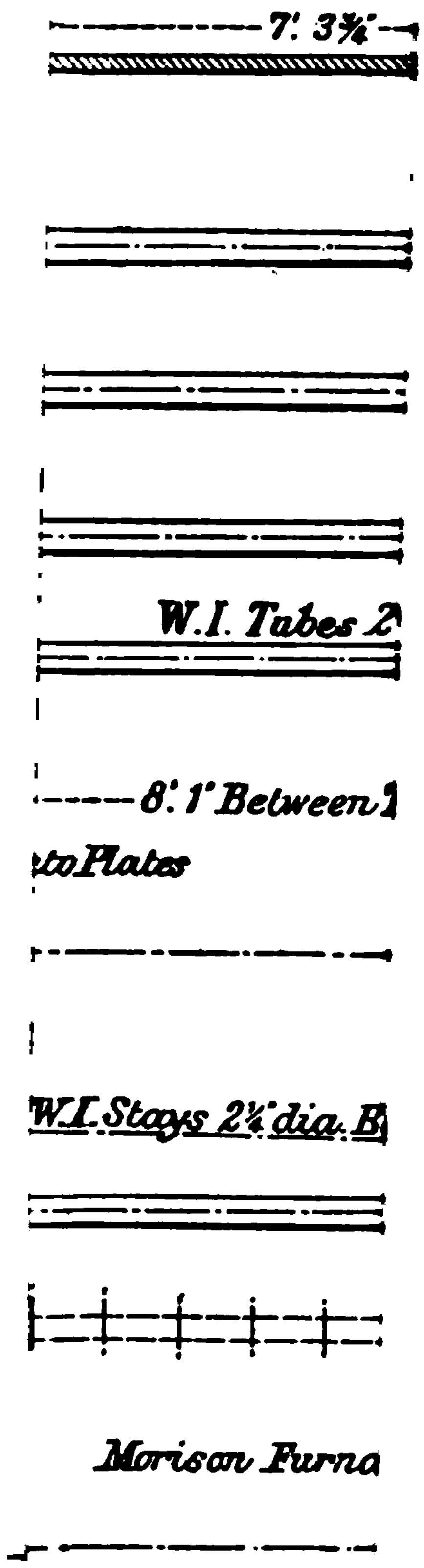
At the sides the boiler has sets of inclined tubes arranged slightly different from the centre series, but forming with them the effective heating surfaces of the boiler. The furnace sides are lined with fire-bricks, and the boiler itself is covered in by a light wrought-iron casing which can be removed when it is neceseary to obtain access to the tubes, etc., for repair or cleaning. In this type of boiler the circulation is particularly well provided for, the water rising up through the inclined tubes, past the uptake headers, and into the steam and water drum, and returning by means of the downward headers. The mud drum at the bottom traps the impurities, such as sediment, etc., and these are blown out of the collector by the cocks fitted for that purpose.

The joints are all metal to metal, and in the case of the tubes the ends are simply expanded into the plates, no screwed joints being used.

Expansion of the boiler under heat is allowed for by the manner in which the mud drum is held down to the foundations.

Yarrow Boiler (4).

Perhaps the two best known types of water tube boilers in general marine practice are the Yarrow and the Babcock & Wilcox. The Yarrow boiler is of simple construction, consisting of two bottom water and mud drums, and one top steam drum with straight tubes connecting the top drum to the bottom drums. The top drum is circular and the bottom drums of oval shape. The tubes are expanded into the drums, and also bell-mouthed (an extra precaution) to prevent drawing out of tubes. The tubes are easily cleaned, and do not silt up so easily as the curved tubes in other types of boilers. In some cases two large tubes are led from the top drum to the bottom drums at each side; these tubes are termed downcast tubes, and serve as a return connection from the top to the bottom drums when circulation is going on. These tubes, it may be mentioned, are outside of the boiler casing and are not in contact with heat. A large casing, built up of asbestos-lined plates, is fitted outside the tubes, and doors are fitted to allow of easy cleaning out of soot which gathers at the bottom of the casing. A large grate and combustion chamber is one of the features of the Yarrow boiler. Zinc plates are fitted, as in Scotch boilers, in the upper or steam drum, and also in each of the bottom drums. Yarrow boilers are usually installed in close stokeholds, fans being driven to give air pressure required. It is found that water tube boilers give the best results with an even fire of 6 to 7 inches thick, and level firing. In the upkeep of Yarrow boilers it is necessary to



Loss of po...
of L.P. steam

Then, Clear gain by feed heating = 9·8 per cent. − 3·4 per cent. = 6·4 per cent.

give air pressure required. It is found that water tube driven to give the best results with an even fire of 6 to 7 inches thick, boilers give firing. In the upkeep of Yarrow boilers it is necessary to and level

clean out soot chambers at least every second day, and the space between tubes also requires cleaning as often as is possible, but the soot chambers can be cleaned while boiler is steaming. Furnace doors and ashpit doors are so arranged that, in event of boiler tube bursting, all doors will close, thus confining escaping steam as much as possible. The tubes vary in diameter from $1\frac{1}{8}$ to $1\frac{1}{4}$ inches, the lower two rows being of still larger size.

Note.—During construction a slight "set" downwards is given to the lower two rows of tubes so as to counteract the expansive buckling effect produced by heat which takes place when under steam.

Economy of Feed Water Heating in Reciprocating Practice.

(Weir System.)

Data.—

Boiler Steam, 185 lbs. gauge.
L.P. „ 10 „
Hot well Temperature, 110°.
„ feed „ 220°.
Heating steam drawn from L.P. chest.

Then, $185 + 15 = 200$ **lbs. absolute boiler steam.**

200 lbs. absolute $= 381 \cdot 7$ **temperature (from Table, page 7).**

Total heat $= 1115 + \cdot 3 \times T°$.

„ $= 1115 + \cdot 3 \times 381 \cdot 7 = 1229 \cdot 5$ **B.T.U.**

Again, $10 + 15 = 25$ **lbs. absolute L.P. steam.**

25 lbs. absolute $= 240°$ **temperature (from Table, page 3).**

Total Heat $= 1115 + \cdot 3 \times 240 = 1187$ **B.T.U.**

Heat available for feed heating per lb. L.P. steam $= 1187 - 110° = 1077$ **B.T.U.**

Increase of feed temperature required $= 220° - 110° = 110°$.

Percentage of L.P. steam required for feed heating $= 1077 : 110 :: 100$ **per cent.** $= 10 \cdot 21$ **per cent.**

Heat required per lb. water for evaporation in boiler without feed heating $= 1229 \cdot 5 - 110 = 1119 \cdot 5$ **B.T.U.**

Heat required per lb. water for evaporation in boiler with feed heating $= 1229 \cdot 5 - 220° = 1009 \cdot 5$ **B.T.U.**

Extra water evaporated due to feed heating $=$ **As,** $1119 \cdot 5 : (1119 \cdot 5 - 1009 \cdot 5) :: 100$ **per cent.** $= 9 \cdot 8$ **per cent.**

Then,

Assuming about equal horse-power in each of the three cylinders:—

Loss of power due to withdrawal of L.P. steam $= 10 \cdot 21$ **per cent.** $\div 3 = 3 \cdot 4$ **per cent.**

Then, Clear gain by feed heating $= 9 \cdot 8$ **per cent.** $- 3 \cdot 4$ **per cent.** $= 6 \cdot 4$ **per cent.**

From the foregoing it will be evident that the heat units given up by condensation in the heater of each pound of L.P. steam is sufficient (neglecting transmission losses, etc.), to heat up 9·6 lbs. of feed water from 110° to 220° temperature, as, 1077 ÷ 110° = 9·6 lbs. water heated.

Boiler Efficiency.

The efficiency of a boiler is the ratio of actual to theoretical evaporation of 1 lb. coal for water from and at a temperature of 212°, which latter is expressed as the "equivalent evaporation," and is the standard taken for test purposes.

To obtain "Equivalent Evaporation."

Example. Pressure, 165 lbs. gauge; temperature, 372·8°; feed temperature, 160°; actual evaporation (measured), 8·5 lbs. water per pound coal. Find equivalent evaporation.

Then, $$\text{Equivalent evaporation} = \frac{(1115 + \cdot 3T^{\circ} - t^{\circ}) \times \text{lbs. water}}{966}.$$

Where, T° = Steam temperature, t° = Feed temperature.

So that, $$\frac{(1115 + \cdot 3 \times 372\cdot 8^{\circ} - 160^{\circ}) \times 8\cdot 5}{966} = 9.38 \text{ lbs. (from and at 212).}$$

Observe that 966 B.T.U. $= 1115 + \cdot 3 \times 212^{\circ} - 212^{\circ}$.

Calorific (Heat) Value of Coal.—The following table gives the average composition and heat unit value of Welsh, Newcastle, and Scotch coal, also of petroleum.

COMPOSITION AND HEAT VALUE OF FUEL.

Name.	Carbon.	Hydrogen.	Oxygen.	Sulphur.	Ash, Nitrogen, etc.	Total Heat Units per Pound.
Welsh coal	84	4·8	4	1·5	5·7	14700 B.T.U.
Newcastle coal	82	5·3	0	1·2	5·5	14600 "
Scotch coal	78	6	9	1	6	14000 "
Petroleum	86·5	12·3	1·2	...	...	20000 "

The heat value of the fuel must be known to obtain anything like accurate results, and usually a sample of the coal is specially analysed to obtain the correct calorific value per pound.

Then,
$$\text{Boiler efficiency (per cent).} = \frac{\text{Equivalent evaporation} \times 966 \times 100 \text{ per cent.}}{\text{Calorific value}}.$$

Example 1. Steam pressure (gauge), 170 lbs.: feed temperature, 180°; actual evaporation per pound coal (measured), 8·6 lbs. water;

calorific value of fuel (Scotch coal), 14000 B.T.U. Find boiler efficiency.

Then, $170 + 15 = 185$ lbs. absolute pressure.

And, 185 lbs. $= 375°$.

Also, $1115 + \cdot 3 \times 375° - 180° = 1047\cdot 5$ B.T.U. required for evaporation.

$$\text{Equivalent evaporation} = \frac{1047\cdot 5 \times 8\cdot 6}{966} = 9\cdot 32 \text{ lbs. water from and at } 212°.$$

$$\text{Boiler efficiency} = \frac{9\cdot 32 \times 966 \times 100 \text{ per cent.}}{14000} = 64.3 \text{ per cent.}$$

Example 2. Pressure, 200 lbs.; feed temperature, 170°; actual evaporation (measured), 9 lbs. water per pound coal; calorific value of coal, 12500 B.T.U. Find boiler efficiency.

Then, $200 + 15 = 215$ lbs. absolute.

And, 215 lbs. $= 388°$ temperature, from table, page 116.

$$\text{Equivalent evaporation} = \frac{(1115 + \cdot 3 \times 388° - 170°) \times 9}{966} = 9\cdot 88 \text{ lbs. from and at } 212°.$$

$$\text{Boiler efficiency} = \frac{9\cdot 88 \times 966 \times 100 \text{ per cent.}}{12500} = 76\cdot 3 \text{ per cent.}$$

The efficiency ranges from about 65 to 80 per cent. in ordinary cases, and varies with the class of boiler. The efficiency is increased when the steam is superheated.

Coal and Water Consumption.—In testing a boiler for efficiency the coal is supplied in measured quantities, and the feed water is passed through measuring tanks or barrels, so that accurate records may be obtained of the actual coal burnt and water evaporated in a given time. From this the actual evaporation per pound coal is obtained as follows:—

$$\frac{\text{Total feed water evaporated, say, per hour (as measured)}}{\text{Total coal consumed per hour (as measured)}} = \text{Actual evaporation.}$$

Example. In testing a boiler plant for the efficiency, the coal fired in twenty-four hours (test period) measured 54620 lbs., and the water evaporated in the same time as measured with the tanks was 478500 lbs. Find the actual evaporation per pound.

Then, Actual evaporation $= 478500 \div 54620 = 8\cdot 76$ lbs. Answer.

Exercise 1. Find boiler efficiency, given the following:—Pressure, 180 lbs. gauge; feed water temperature, 162°; actual evaporation, 8·8 lbs. water; and heat value of fuel, 13000 B.T.U. per pound. Answer. 72 per cent.

Exercise 2. Find boiler efficiency, given the following:—Pressure, 215 lbs. gauge; feed water, 190° temperature; total coal burnt per hour, 2500 lbs.; and total water evaporated per hour, 21500 lbs. Heat value of coal 13500 B.T.U. per pound. Answer. 66·4 per cent.

Superheated Steam.—If the steam is superheated the number of heat units required to produce the rise of superheat temperature

must be added to the total heat of the steam. To obtain the number of heat units, the specific heat constant of superheated steam at constant pressure has to be used, and this is usually taken at ·48, which means that to raise one pound of steam 1° Fahr. requires only ·48 of the heat required to raise one pound of water 1° Fahr.

Therefore, $\text{Heat units} = \cdot 48 \times \text{Degrees superheat.}$

Example. Find number of heat units added to steam to produce a superheat of 100° Fahr. (from 366° to 466° Fahr.).

Then, $\text{Heat units} = \cdot 48 \times 100° = 48 \text{ B.T.U.}$ Ans.

Equivalent Evaporation (with superheated steam).

$$\frac{[(1115 + \cdot 3 \times T° - t°) + \cdot 48 \times (T_1° - T°)] \times \text{Pounds water}}{966}.$$

Where, $T°$ = Boiler steam temperature.

" $t°$ = Feed temperature.

" $T_1°$ = Temperature of superheated steam.

" 966 = Latent heat of evaporation from and at 212°.

Example. Boiler pressure, 200 lbs. gauge, superheated to 500° Fahr. Feed temperature, 190°; actual evaporation (measured) 9·3 lbs. water per pound coal; and calorific value of coal, 13000 B.T.U. per pound. Find boiler efficiency.

Then, $200 + 15 = 215$ lbs. absolute.

Temperature = 388°, from table, page 116.

Therefore,

$$\left.\begin{matrix}\text{Equivalent}\\ \text{evaporation}\end{matrix}\right\} = \frac{[(1115 + \cdot 3 \times 388° - 190° + \cdot 48 \times (500° - 388°)] \times 9\cdot 3}{966}$$

$$= \frac{(1115 + 116\cdot 4 - 190 + 53\cdot 76) \times 9\cdot 3}{966}$$

$$= \frac{1095\cdot 16 \times 9\cdot 3}{966} = 10\cdot 45 \text{ lbs. water from and at } 212°.$$

$$\text{Boiler efficiency} = \frac{10\cdot 43 \times 966 \times 100 \text{ per cent.}}{13000} = 77\cdot 6 \text{ per cent.} \quad \text{Ans.}$$

Exercise. Boiler pressure, 185 lbs. gauge superheated up to 510° Fahr.; feed temperature, 165°; actual evaporation, 9 lbs. water per pound coal; and heat value of coal, 13500 B.T.U. per pound. Find boiler efficiency. Answer. 74·9 per cent.

Superheated Steam.

Of late years a decided reaction has set in among marine engineers in favour of superheated steam, which, as proved conclusively by exhaustive experiments, possesses undoubted practical advantages, and the use of which results in considerable economy. It has been found that with superheating to the extent of 50° the gain is about 5 per cent., and if the superheat is increased to 200° above the

natural pressure temperature of the steam, then the resulting economy is about 12 per cent.

It should be noted that if saturated steam—that is, steam drawn from a boiler—is raised in temperature and the *pressure* kept constant, the volume increases; this means a larger volume of steam produced for the same amount of water evaporated, and therefore less boiler space required. Again, by superheating steam which originally contains water, the water is evaporated, thus giving drier steam, which results in reduced cylinder condensation losses, and less transfer of heat to the cylinder or turbine casing walls. It will thus be seen that the advantages of superheated steam are beyond question.

NOTE.—The volume varies with the absolute temperature if the pressure is kept constant.

EXAMPLE.—Boiler steam (saturated) at a pressure of 180 lbs. gauge pressure, temperature of 380°, and specific volume 2·31 cubic feet; find the volume if the steam is superheated 100° Fahr.

Then, $380° + 100 = 480°$ **steam temperature when superheated.**

And, $461 =$ **absolute temperature constant.**

Therefore, as, $(380° + 461°) : (480° + 461°) :: 2{\cdot}31 = 2{\cdot}58$ **cubic feet volume.**

So that the volume of the steam per pound is now increased by $2{\cdot}58 - 2{\cdot}31 = {\cdot}27$ **of a cubic foot.**

In addition to this it must be remembered that the temperature is higher, and the steam of a drier condition, owing to freedom from water particles.

Advantages of Superheated Steam.

1. Increase of steam volume.
2. Dry steam enters the turbines, and reduced condensation losses result.
3. Less leakage of steam past valves and pistons.
4. Less danger from water hammer in main steam pipes or chests.
5. Reduced erosion of turbine blades.

Against these gains there are, however, certain disadvantages, which also require to be taken into account.

Disadvantages of Superheated Steam.

1. Difficulty of lubrication.
2. Piston rings more easily broken.

} Reciprocating Practice.

3. Danger of blade stripping due to excessive expansion, more particularly in the case of astern turbines when reversing.
4. Initial stage blading, etc., to be of copper to withstand the effects of superheat.

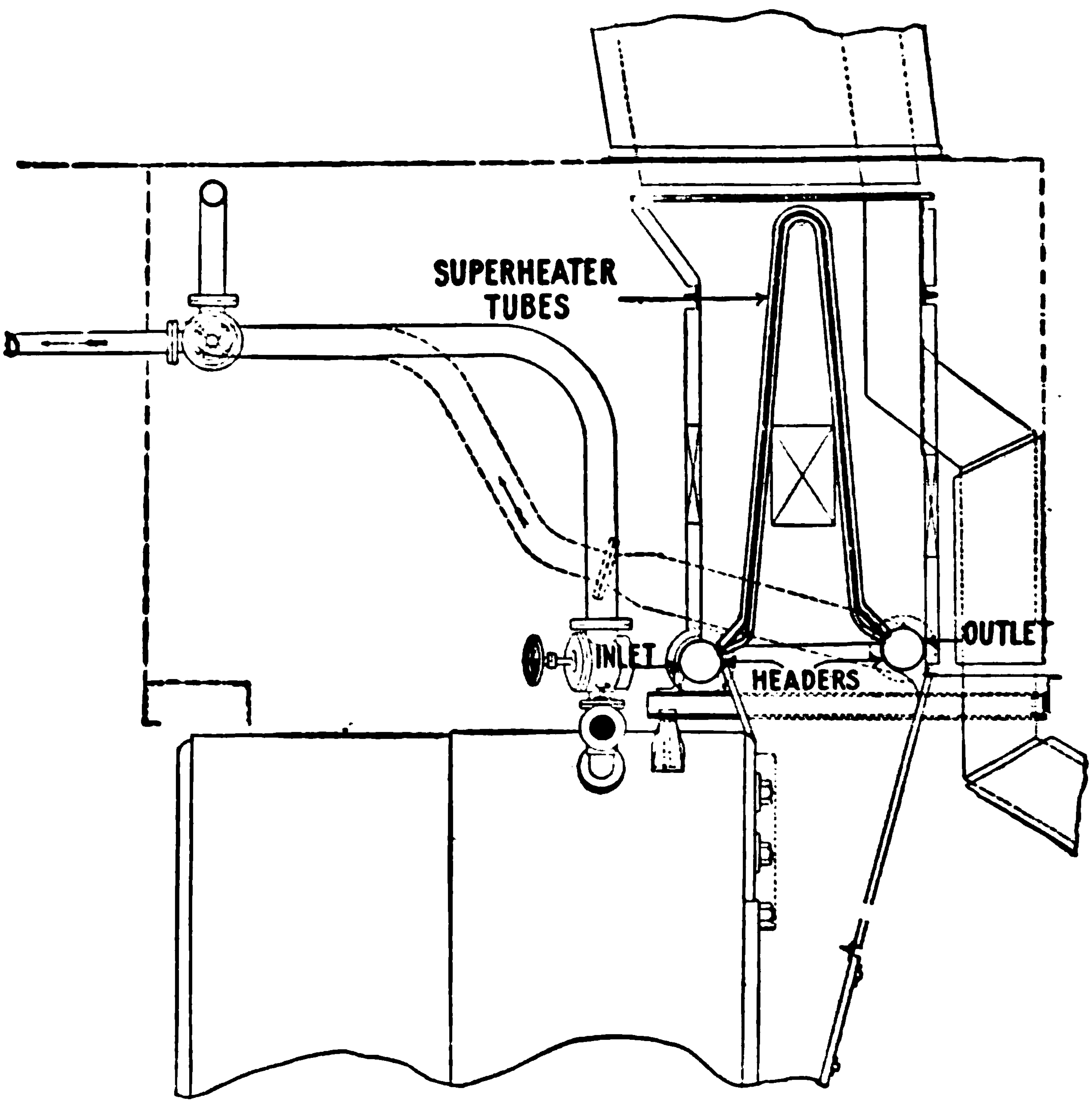
} Turbine Practice.

5. Steam piping, and all valves and seats, etc., through which superheated steam passes, require to be of steel.

6. Trouble experienced in keeping superheater coils or tubes tight and in good working order, this constituting the more serious practical disadvantage.

7. If boilers prime, danger of closing up of superheater tubes due to deposit of soluble matter carried over with the priming water.

Methods of Superheating.—In marine practice the ordinary method has been to utilise the funnel gases in superheating the steam,



No. 5.—"Watkinson" Type Marine Superheater.

although the independently fired type as devised by Professor Watkinson has also been fitted in some steamers.

The Watkinson superheater (Sketch No. 5) consists of a series of U-shaped mild steel tubes, expanded into large headers similar to those employed in water-tube boilers. This nest of tubes is fitted in the uptake, and the steam from the boiler enters one of the headers, and passing through the U-shaped tubes becomes superheated by the furnace gases which are flowing over the tubes. The steam then

enters the other header and flows along the steam pipe to the engine. Suitable drainage arrangements are fitted to keep the drums and tubes clear of water, and by suitable bye-pass valves and pipes the steam may, if required, pass direct from the boilers to the engine, without entering the superheater coils.

When it is stated that the average loss due to initial cylinder condensation in reciprocating practice is about 15 per cent. with saturated steam, the advantage of superheated or dry steam will be apparent, as less water being present in the steam the condensation losses are greatly reduced, and may be practically eliminated.

To overcome the lubrication difficulty, special high temperature mineral oils are now manufactured by the various oil companies, which are said to resist the disintegrating effects of the superheated steam, and allow of suitable lubrication of rods, pistons, and valves.

Regarding the resulting increase of volume due to superheating, Professor Watkinson says: "During superheating, although the pressure of the steam remains constant, its volume is greatly increased. The amount of heat required to superheat 1 lb. of steam by 150° Fahr. is 72 British heat units, which is only about 6 per cent. of the heat required to generate 1 lb. of dry saturated steam. The increase in volume due to this additional 6 per cent. of heat averages about 30 per cent."*

Schmidt or Robinson Type Superheater (6-7).

For marine practice these types of smoke-box superheaters have proved fairly satisfactory, and have recently been fitted in the boilers of quite a large number of new vessels, including many supplied with geared-down turbines, in which a moderate degree of superheat (say from 100° to 150°) at engine room stop valve is found sufficient.

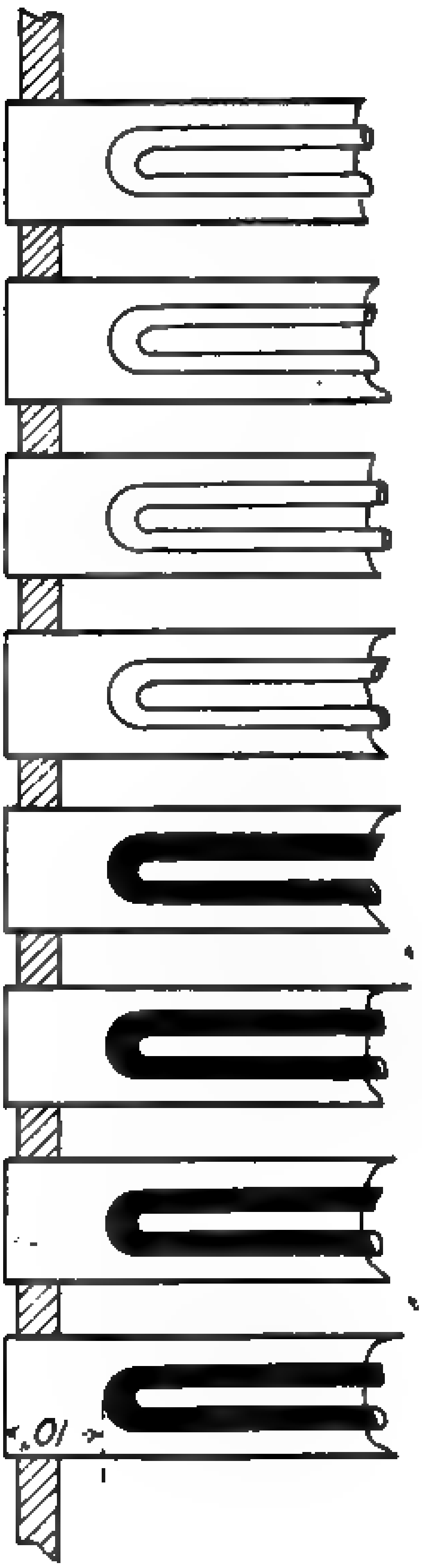
The following data of superheat working is taken from a large quadruple expansion engine set, and the results obtained in this case were very satisfactory indeed, the economy of superheated steam over saturated steam showing clearly on the coal consumption :—

Superheat Data.

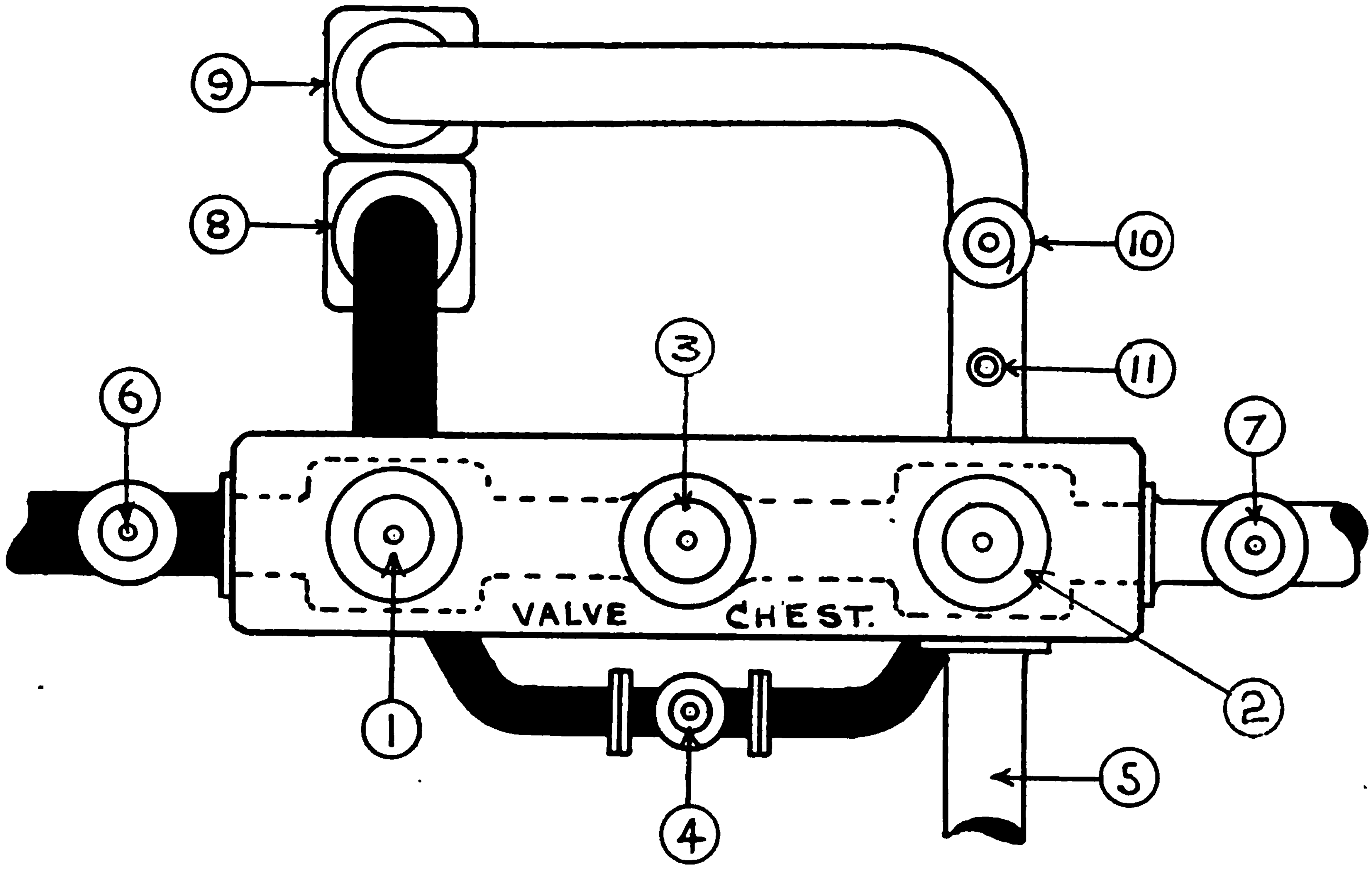
Type of engines	Quadruple expansion.
I.H.P.	4200
Boilers	4 single-ended.
Diameter of smoke tubes	3″
Boiler pressure	220 lbs. gauge.
Diameter of superheater tubes	$\frac{3}{4}$″
Draught	Howden forced.
Air pressure at fan	$2\frac{3}{4}$″
„ under bars	$\frac{3}{4}$″
„ above „	$\frac{3}{8}$″
Revolutions of fan	260 p.m.

*Specific heat of superheated steam = ·48.

Therefore, $150° \times \cdot 48 = 72$ B.T.U.



No. 6.—Schmidt Marine Type Superheater.



No. 7.—Schmidt Superheater Valve Chest.
(Fitted on Top of Boilers.)

(1) Main stop valve giving steam from boilers either to superheater header box, or to engines direct through bye-pass valve (3).

(2) Superheater steam stop valve to engines.

(3) Bye-pass valve, for direct boiler steam to engines, through (1), (3), (2), and (5).

(4) Mixing valve for giving a mixture of boiler steam and superheated steam as may be found necessary.

(5) Main steam pipe to engines (either boiler steam or superheated steam).

(6) Auxiliary boiler steam stop valve.

(7) ,, superheated ,, ,, ,,

(8) Boiler steam header (before superheat).

(9) Superheat ,, ,, (after ,,

(10) Safety valve (single) which may be required if valve (2) is shut and steam is being passed through valve (1) and bye-pass (3), that is, the superheater shut off from boilers.

(11) Fitting for connecting up pyrometer (high temperature recorder).

NOTE.—With superheater off when working bye-pass, it is advisable to have main stop valve (1) eased off seat to allow a little moisture to be present in the U-tubes of the superheater, otherwise damage to the tubes, etc., by dry steam at high temperature is likely to take place.

DATA OF PRESSURES AND TEMPERATURES WITH SUPERHEAT (RECIPROCATING PRACTICE).

Position.	Gauge Pressure in Lbs. per Sq. In.	Temperature for Pressure.	Temperature as Tested.	Degree of Superheat.
Boiler steam	220	396°	590°	194°
H.P. steam	210	392°	580°	188°
1st I.P. steam	110	344°	440°	96°
2nd I.P. steam	47	295°	295°	0°
L.P. steam	9·5	236°	230°	—6°

Smoke-box temperature	385°
Uptake temperature	370°
Funnel ,,	355°

It should be noted that the superheating of the steam by the waste gases extracts the heat of the latter, and lowers the funnel gas temperature, as shown by the recorded results; this again has the effect of reducing the amount of heat available for the air heating tubes of the forced draught system.

The U-tubes extend to within less than a foot of the back end then bend round and return again, again bend round, and enter another tube, &c.

One complete element is shown in black section, the other element is left open.

As the tubes incline to block up with soot, "diamond" blowers or other tube-clearing appliances are employed at regular intervals to keep them clear.

Working Parts.

With high degrees of superheat the working parts, such as main steam pipe, stop valves, nozzle valves, control boxes, etc., require to be of steel, and the initial rows of blading (if reaction turbines) of copper. As, however, nearly all reaction turbines are now fitted with an impulse velocity wheel, at the first stage this difficulty is overcome, as only steam of about half the initial pressure and of lower temperature passes through the first rows of reaction blading, so that the temperature is not excessive.

As superheating drys the steam, the water particles present in saturated steam are absent, and the effect of this on the wear of the turbine blading (erosion) is much less than with direct boiler steam.

Practical Difficulties.

The chief objection to superheating has always been, is at present, and apparently always will be, the difficulty of obtaining a reliable type of superheater which will be able to resist the severe

conditions of temperature to which it is exposed. In many cases the smoke-box type gives out by leakage of the superheater tubes at the header joints, and it is quite a common experience for a vessel to leave port with, say, all superheaters in operation, and after a few weeks it may be found necessary to cut out a number of them owing to serious leakage at the header connections, so that, after all, the turbines require to be driven ultimately by direct steam.

Superheat and Reversing.

For sudden stopping and reverse running superheated steam is understood to be unsuitable, owing to the high temperature produced in the astern blading by the heavy strain in overcoming the ahead way of the vessel, combined with the initial high temperature of the steam, which may result in blade stripping by abnormal expansion of the blades in the case of reaction blading. With a velocity wheel, however, the risk is not so great, and the present practice is to make the astern turbine of the impulse velocity wheel type without any reaction blading whatever, even when reaction ahead turbines are fitted. For this reason, with reaction blading, it is usual to only admit steam of from 80 lbs. to 100 lbs. pressure to the astern turbines, the drop in pressure being obtained by throttling.

Again, if superheat only is used for all steam purposes, the auxiliary machinery requires to be specially designed of steel to meet the conditions imposed by the high temperature and dry nature of the steam, which is most searching at joints, and plays havoc with piston rings, rods, valve faces, etc.

SECTION XVII.

MARINE TURBO-ELECTRIC DRIVES.

The arrangement of the marine turbo-electric drive is usually as follows:—

1. Turbine drive to electric generators.

2. Generator drive to electric motors.

3. Motor drive through single reduction gearing to propeller shaft.

As the generators run at, say, 3000 revs., and the motors 500 revs. this constitutes the first step in speed reduction, the single gearing forming the second step, and bringing down the shaft-speed to 85 or 90 revs. per minute as best suited for maximum propeller efficiency.

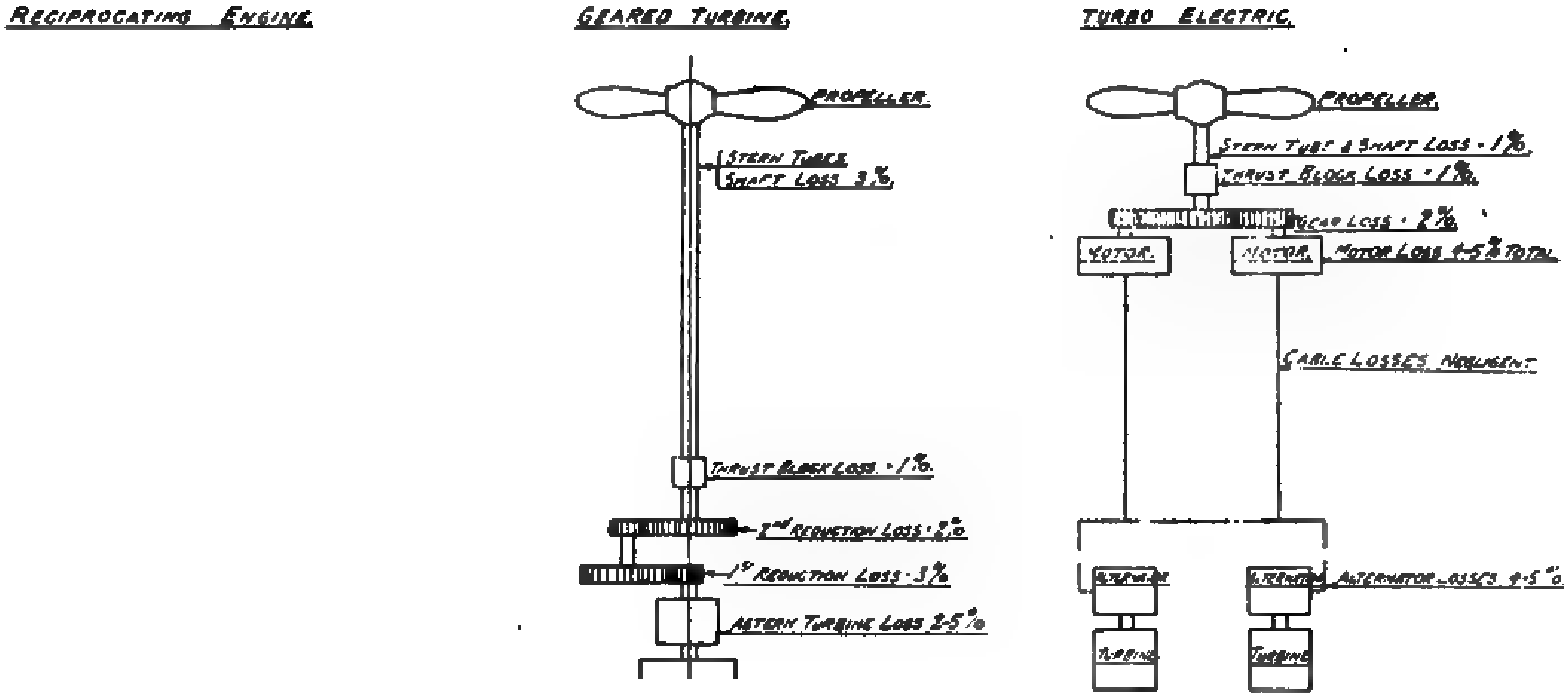
In this system of marine propulsion one of the principal advantages lies in the saving in machinery space, which gives a corresponding additional cargo or passenger space in the vessel.

Again, the electrical gear (against which considerable prejudice still exists among many marine engineers), is of a comparatively simple type, and of strong construction, correctly described as "fool proof" in every way, the whole of the controlling gear for starting, stopping, ahead and astern, being contained in a box fitted with all the necessary appliances for easy and accurate control, in addition to the various safety and emergency devices specially designed for reliability, ease of manipulation, and security from breakdown under the severe conditions of marine practice.

The line shafting is pratically eliminated, as only a short propeller length is required to couple up the tail shaft gear wheel to the electric motor pinions. Then as regards running astern, the whole power can be developed, when reversing, in place of about only 60 per cent., which is the usual proportion in turbine practice.

Transmission Losses.

The combined losses by power transmission from (1) turbine to generator, (2) generator to motors, (3) motors to shaft, appear to average about 14 or 15 per cent., varying to a certain extent with the power transmitted.



No. 1a.—Diagram of Comparative Transmission Losses—Reciprocating, Geared Turbines, and Turbo-Electric.

Reciprocating Engine Losses = 11 + 1 + 3 = 15 per cent. total.
Geared Down Turbine ,, = 2·5 + 3 + 2 + 1 + 3 = 11·5 per cent. total.
Turbo Electric ,, = 4·5 + 4·5 + 2 + 1 + 1 = 13 per cent. total

Generators and Motors.

"The generators invariably develop 3-phase current at a pressure of about 500 volts in the case of moderate powers up to 2500 S.H.P. per vessel. In the case of larger ships of, say, 10000 S.H.P. and upwards, where the motors are placed aft, and the generator station amidships, the voltage generated would be of the nature of 3000 upwards; in the case of twin-screw vessels of 10000 S.H.P. and over, direct-coupled motors are generally employed in place of geared motors. Direct-coupled motors will be used in the new American battleships."

In the Ljungstrom type two generators are coupled to the turbine, and, as in other makes, two motors connect through pinions to the driving shaft gear wheel.

In other types, such as the British Thomson-Houston-Curtis turbo-turbine drive system, one generator only is fitted per shaft, with a pair of motors geared down to the propeller length of shafting.

Condensers.

The condensers are generally placed directly underneath the turbine, as this arrangement allows of efficient drainage of the latter in addition to an important saving in machinery space.

Pumps, etc.

The pumps, such as air, circulating, bilge, feed tank, etc., are either driven direct by steam, or are arranged with electric motor drives, this latter system being generally preferred by the builders.

Feed Heaters and Evaporators.

The steam required for feed heaters and evaporators is tapped from some point in the turbine system where pressures are most economical for the purpose.

Steering Gear, etc.

The steering gear, deck winches, and windless gears, etc., are either driven by steam, or by electrical motor drives, fed from the main generators or from an auxiliary set.

Kilowatts and B.H.P. or S.H.P.

A kilowatt = 1000 watts, and a horse-power 746 watts.

Therefore, 1 kilowatt = 1·34 H.P. as 1000 ÷ 746 = 1·34.

To Convert Kilowatts to S.H.P.

RULE.—Multiply the kilowatts by 1·34.

EXAMPLE 1.—A turbo-electric drive turbine gives out 1500 kilowatts. Express this in B.H.P.

Then, $1500 \times 1{\cdot}34 = 2010$ B.H.P.

EXAMPLE 2.—The steam consumption of a turbine coupled to an electric generator is 14·5 lbs. per hour. Express this per S.H.P. per hour.

Then, $12{\cdot}5 \div 1{\cdot}34 = 9{\cdot}3$ **lbs. steam per S.H.P. hour.**

EXAMPLE 3.—A turbo-electric drive develops 1800 S.H.P. Find the output of the generator in watts.

Then, $1800 \div 1{\cdot}34 = 1343$ **watts.**

Through the joint courtesy of R. S. Portham, Esq., of the British Marine Ljungstrom Co. Ltd., and the Directors of the Brush Electrical Co. Ltd., the writer was enabled to pay a visit to the works of the latter firm at Loughbrough to view turbine sets under construction, and others assembled complete. After careful examination of the various details which make up this turbine, the conclusions arrived at verify the claims made by the builders as regards strength of blading, accuracy of the work in general, and reliability under running conditions of the Ljungstrom turbine, which, it may be mentioned, strikesthe observer as being without doubt the smallest machine of its class for any given power output.

For this type of turbine the steam (or coal) consumption per kilowatt or per S.H.P. per hour, works out to a figure which a few years ago would have been deemed impossible, actual results from practice giving less than one pound coal per H.P. per hour, which may be taken as about the last word in turbine efficiency up to the present date.

Being of the reaction radial flow design the blade rings as fitted on the face discs are stiffened and strengthened in a manner which, after close inspection, leaves no doubt as to the existence of a good margin of strength when under maximum conditions of work. All of the other details of this "dwarf" type of turbine also bear evidences of originality of design, previously accepted standards having been abandoned to give place to the development of newer ideas.

The running control gear appears to be arranged to allow for any and every possible emergency, and danger of total breakdown is practically eliminated, while the time required in stopping and changing from ahead to astern gear is merely a matter of seconds.

For a description of the action, construction, and general data of the Ljungstrom turbine, the writer cannot do better than give a reprint of the excellent paper delivered by R. S. Portham, Esq., before the Members of the Institution of Engineers and Shipbuilders in Scotland on the 21st March 1916.

The Ljungstrom Turbine and its Application to Marine Propulsion.*

By ROLAND S. PORTHAM, Esq.

At the end of the nineteenth century the marine engineer's task in the selection of propelling machinery was a comparatively easy one.

* Read before meeting of Institution of Engineers and Shipbuilders in Scotland on 21st March 1916.

Practically his only alternatives were the installation of either triple or quadruple-expansion steam engines. Within the last few years, however, the range open to him has been much wider; and although the reciprocating steam engine still has many adherents, the direct drive turbine, either independently or in conjunction with a reciprocator, finds advocates for large steamers with three or four shafts. The mechanically-geared turbine, either impulse or impulse reaction, has rapidly come into favour for many types of vessels, and the internal-combustion engine has a school of its own, whilst recently turbo-electrical propulsion has come to the front.

It is a far cry from the first electrically-propelled 28-feet boat designed by Prof. Jacobi, which ran on the Neva in 1838, driven from a primary battery, to the 37000 S.H.P., 32000-ton, 22½-knot, super-dreadnought, "New Mexico" (late "California"), the keel of which was laid in October 1915. The march of progress in electrical marine propulsion, from 1838 to within the last two years, has been very slow, and the recent forward movement is remarkable and significant.

In 1903, Messrs Nobel Bros. ran a tank vessel of 1150 tons displacement on the Volga, the vessel's speed being about eight knots, and the total of 360 B.H.P. was supplied by three sets of Diesel engines to three motors, each driving a separate propeller. I believe three or four other similar Diesel electrically-propelled ships were built for Messrs Nobel from 1906 to 1908, and are still in operation.

About 1912, the late Mr Henry Mavor's firm equipped a Canadian Lake steamer—the "Tynemount"—with two 250 B.H.P. Diesel alternator sets, supplying current to a three-phase motor. This was the first example of three-phase machinery applied to marine propulsion, but was unsuccessful, owing to the difficulty experienced in running the two Diesel engines at the lower speeds.

The first practical example of turbo-electrical transmission was on the United States collier "Jupiter," of 20000 tons displacement, and 12000 tons dead weight, equipped with one 5000 K.W. turbo-generator of the Curtis type, driving two direct-coupled three-phase motors, each of 2500 B.H.P. The successful operation of this ship for more than 2½ years has resulted in orders being placed by the United States naval authorities for three electrically-propelled super-dreadnoughts, of which the "New Mexico" already referred to is one.

Early in 1913, the favourable results achieved by the turbine associated with his name led Mr Birger Ljungström, of Stockholm, in conjunction with his brother—Mr Frederick Ljungström—to seriously consider the question of electrical marine-propulsion for commercial vessels.

The pioneer ship built—the "Mjölner"—of 900 S.H.P., with two generating units, each of 450 S.H.P., has successfully proved Messrs Ljungström's contention with regard to the economical adaptation of this system, even for one of the smallest sizes of sea-going ships. This vessel has now been running since December 1914, and the fuel consumption under the same conditions of dead weight, speed, and coal has averaged 38·6 per cent. less than that of her sister ship—

"Mimer"—fitted with triple-expansion reciprocating engines. A short description of the installation on this ship and the relative trials are given at the end of the paper.

Messrs Ljungström recognised that an important feature in marine propulsion, and one not hitherto sufficiently recognised, was that the generating plant should be subdivided into at least two entirely independent units, so that either set could drive the ship at about three-quarters speed, running under relatively economical conditions. Naturally, this subdivision of power, though conducing to maximum reliability at sea, necessitates an efficient, light, and accessible turbine.

In 1907, Messrs Ljungström, in the course of investigations on the Curtis impulse and Parsons reaction types of turbines, came to the conclusion, which has been confirmed by independent investigators, that the pure reaction principle offered certain advantages over the impulse type, more especially for turbines of larger output.

The drawback, however, in the reaction system was the large number of rings and blades, involving a correspondingly long rotor to ensure a reasonable efficiency, and the attendant tendency to distortion of casing with varying temperatures, necessitating considerable tip clearances and corresponding leakages.

With a view to reducing such excessive length, Messrs Ljungström experimented on various widths of blades, and found that these could be reduced to 5 mm. in the high pressure end up to 17 mm. in the low pressure end without affecting the efficiency.

They realised, however, that shortening the turbine axially did not solve the problem of so adjusting the passage of steam through the blading as to allow it to follow its natural curve of expansion from a stop-valve volume of about 2½ cubic feet per lb. up to a volume of over 300 cubic feet per lb., at a vacuum of from 28 to 28½ inches. They, therefore, conceived the idea of turning the plane of the blade area through an angle of 90 degrees, thus converting the direction of steam from axial or parallel to radial flow, and were thus enabled to proportion the area of the successive blade rings so as to follow the true expansion curve of steam from turbine inlet to condenser.

Simultaneously the great advantages were realised which would accrue were the guide blades in the ordinary turbine designed so as to revolve at the same speed as the moving blades, thus doubling the relative speed of the blade rings, and, therefore, necessitating only one-quarter the total number of rings for the same efficiency.

Earlier inventors, notably Siemens-Schuckert, had attempted revolving the fixed guide-blades in the axial flow turbine by means of bevel gearings. Obviously such devices were unsatisfactory, and Messrs Ljungström concluded that the only satisfactory method of double rotation would be on the pure reaction principle of alternate concentric blade-rings. The steam reacting from the first blade ring to the next ring surrounding and concentric thereto.

Thus there is an entire absence of vibration in running up to full speed. The weight of the turbo-alternator is $15\frac{1}{2}$ tons.

. . . rings. The steam reacting from the first blade ring
to the next ring surrounding and concentric thereto.

As it was important that the clearance between the revolving rings should be as small as possible, the next step was to so hold them in their supporting discs that any distortion consequent on the variation in temperature between the blade ring and the framework should be eliminated. The original idea was, therefore, conceived of carrying each blade ring on its own expansion ring, and embodying a series of expansion or breathing rings throughout all parts of the turbine subject to temperature differences.

The period between 1907 and 1909 was spent by Messrs Ljungström in preparing the designs of all the essential details, and the results of their careful work were embodied, in 1910, in the first turbine constructed by them of 750 B.H.P., running at 3700 revs. per minute, which gave a trial steam consumption of 8·9 lbs. per

No. 2.

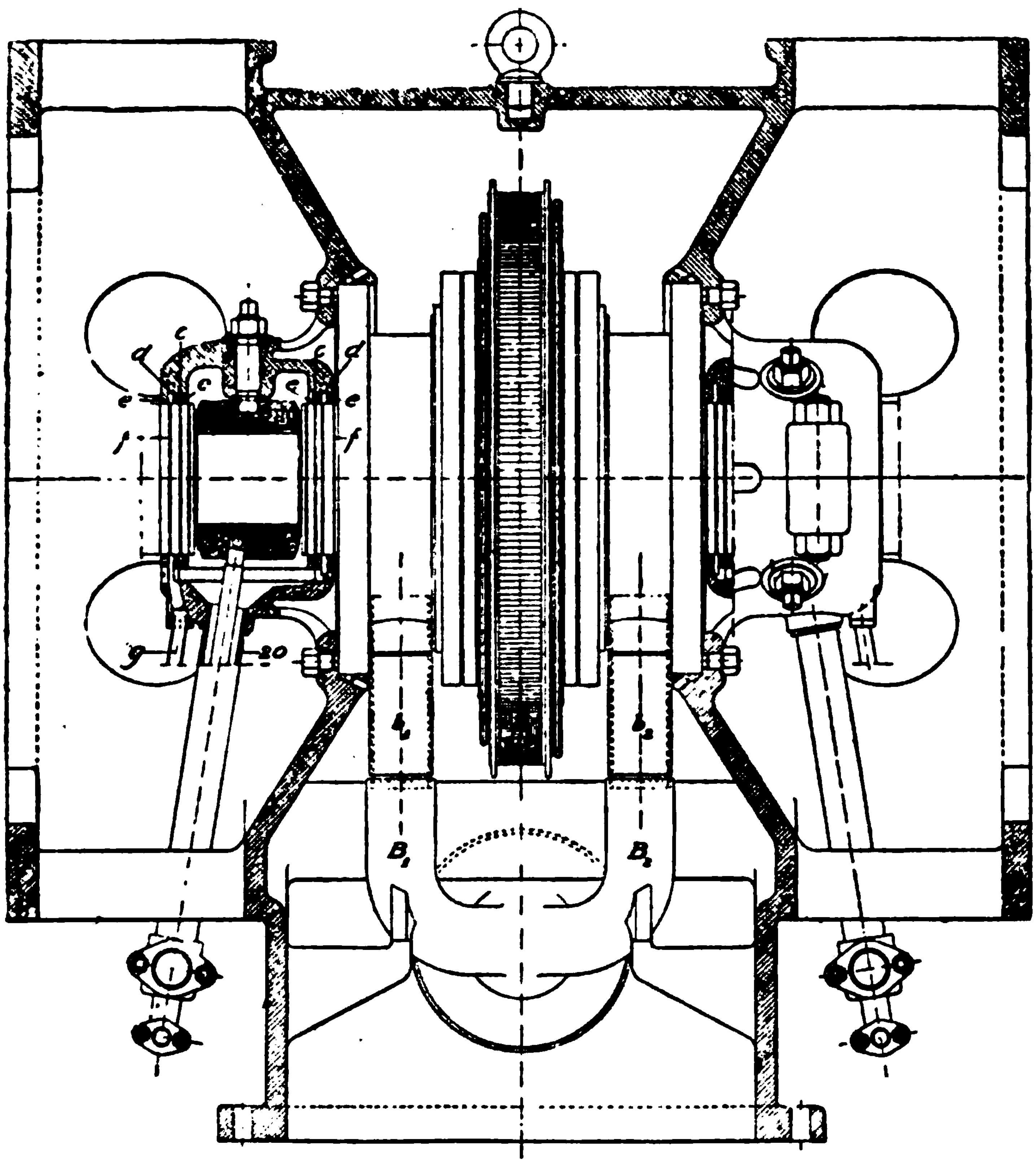
B.H.P. per hour, total temperature of steam 626° Fah., vacuum 2·86 inches, and 171 lbs. per square inch absolute pressure on the turbine inlet-valve.

Since that date an aggregate of approximately 200,000 B.H.P., of a design practically identical to their first turbine, is in operation or under construction.

No. 2 shows the general outside appearance of a 1500 B.H.P. turbo-alternator; the over-all length being 16 ft., and outside diameter 4 ft. 2 in. Two three-phase alternators, each of half the capacity of the turbine, are coupled to opposite ends of the turbine, and run in opposite directions, the speed of the turbo-alternator is 3000 revs. per minute, corresponding to a relative blade speed of 6000 revs. per minute. The running speed is half the critical speed. Thus there is an entire absence of vibration in running up to full speed. The weight of the turbo-alternator is $15\frac{1}{2}$ tons.

No. 3

No. 3 shows a half longitudinal section of the turbine. The turbine revolves on an axis represented by XY. The steam enters through the main steam pipe, branching off where it enters the turbine into two vertical pipes AA, from whence it passes by means of a number of holes BB milled in the hub of each supporting disc to the innermost turbine ring 1, then radially through the two series of blade rings to the last ring 42 on its way to the condenser. There

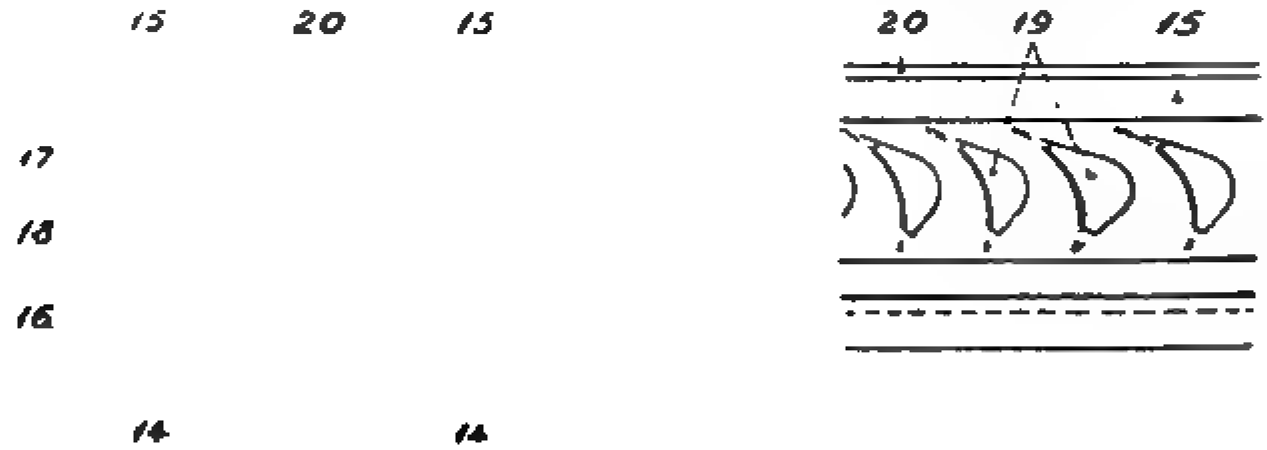


No. 4.

are altogether 42 blade rings, 21 being fixed to supporting disc GR, and 21 to the supporting disc GL, the discs revolving in opposite directions. No. 4 shows an outline arrangement of the turbine, and illustrates a general view of the steam piping.

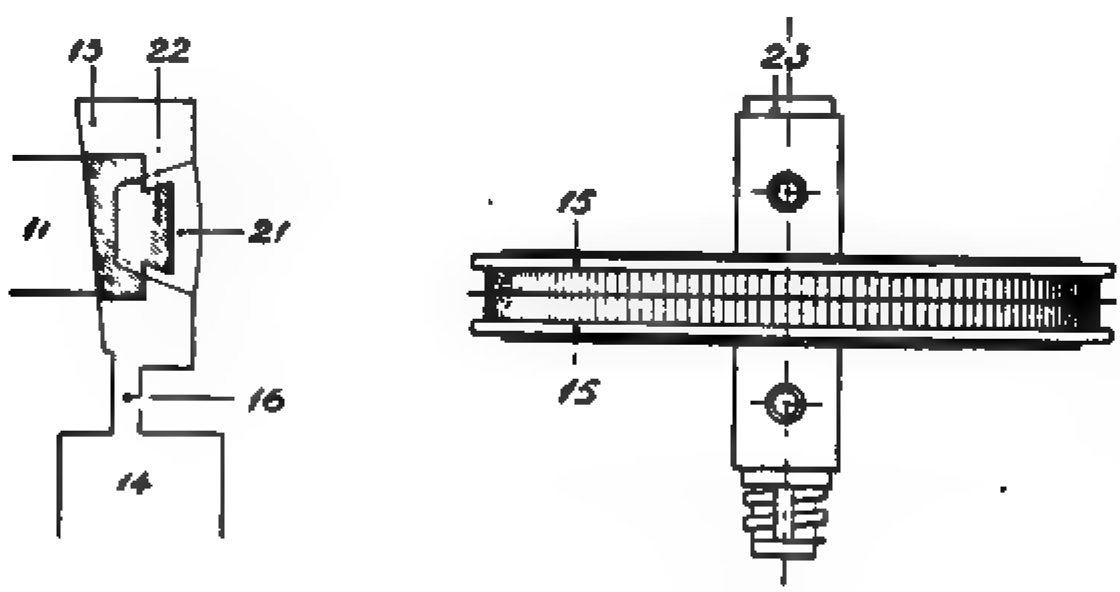
It will be noted that these two supporting discs, No. 3, carrying the blade rings, are each divided into three sections, connected through expansion rings, in order to avoid the stresses and alterations in shape which would arise in a solid disc subjected to the influence

of varying and irregularly distributed temperatures, as would be the case when steam is first admitted into the turbine, and when variations occur in steam pressure, steam temperature, or load. Considerable temperature stresses would occur were they built in solid form, as the centre of the disc is brought into contact with high pressure



No. 5.

steam, while at the circumference it is in contact with low pressure steam of correspondingly lower temperature. The supporting discs are made from chrome nickel steel.



No. 6.

Each disc is directly coupled to the rotor of the generator, being mounted on the overhanging end of the shaft. The centre hole is tapering, and the hub is secured to the shaft by means of a number of round keys, which are in their turn retained by a locking device screwed into the end of the shaft. The part of the shaft situated within the turbine casing, and subjected to heating by the steam, is hollow, so that the fluctuations of temperature in shaft and hub

follow one another, and thus avoid play between the hub and the shaft.

A general inspection of this half-longitudinal section shows a number of small dumb-bell-shaped sections. These are not links, but cross-sections of circular rings made of nickel steel. They are the expansion or breathing rings previously referred to, which form one of the essential features of the turbine, and are interposed between any two parts which would be otherwise liable to distortion due to variation of temperature. These expansion rings are interposed between the blade rings and supporting discs, between the component parts of the discs themselves, and between the steam chests and their supporting frames.

The blade rings of the turbine form an important feature, and Nos. 5, 6, and 7 show the method of their construction. The blades themselves are made of 5 per cent. nickel steel. Calculating from the centre line, XY, No. 3, the first 33 rows are 5 mm. wide, the next five

1

2

3

No. 7.

are 7 mm., and the last two 12 mm. and 20 mm. respectively. The blades are milled from solid rough-turned bars on ordinary horizontal milling machines, and are polished to a mirror-like surface both inside and out. They are then cut into lengths (11), No. 5, and are notched on the ends for insertion into mild steel retaining frames (15), No. 5. A side view of such retaining frames is shown with registers or slots to receive the blades in (19), No. 5. The turbine blades are retained in their positions by means of thin sheet-iron strips (20), No. 5, in which are punched holes of the same cross-section as the blades, and at a distance corresponding to the pitch or spacing of the blades. These strips are applied in their places simultaneously with the assembling of the blades in the retaining frames, the angles of the blades being thus properly set, and the blades prevented from shifting during the welding operation. The discs containing the blades are mounted on a mandril, No. 6, which is held vertically. The blade ends are welded up solid in their frames by melting iron wire under the oxy-acetylene blow-pipe, thus filling up the channels (18), No. 5. The blade ring thus obtained is then turned up in a lathe, the sheet

iron strips are cut away, and the supporting mandril removed, giving the dovetail shape to (22), No. 6, or (10), No. 7. Strengthening rings (9), No. 7, made of the finest grade spring-steel are then dovetailed into the blade ring, and the edges squeezed in so as to grip the dovetail tightly, by means of hinged rollers carried on the lathe tool rest. A groove is formed in one of the s
receive the dumb-bell-shaped expan
end of the dumb-bell being rolled i
which is subsequently fixed to the
secured by a caulking strip (3), No. ;
plete, with the exception of two pure
caulked into the top surface of the st
of soft iron wire (8), No. 7. The fi
prevent side leakage from any pair o

No. 8 illustrates the completed se
tion in the supporting disc.

Referring again to No. 3, it will
ing the centre and flowing through
condenser, will displace each rotating
to the other, unless this is countera
pressure. This is effected by a serie
which D is attached to its respect
expansion rings, and revolves with
stationary part of the steam chest. S
dummies from the centre of the
packings receive the full steam p
decreasing from the centre until it
denser outside of the outermost lab
dummies has a small groove on its
strip of thin nickel is lightly driven.

These labyrinthic dummy packing
method of balancing the axial thru
the turbine, the revolving labyrinth
face E, thereby increasing the space
causing high pressure stream to ent
in the opposite direction, and auton
openings. A small dashpot is fixed
for the purpose of damping any osc
otherwise occur. Reference may be
or glands F, No. 3, which reduce shaft leakage to a minimum. These packings are of nickel steel, designed to expand freely in all directions without distortion. Packing for a 1500 B.H.P. turbine occupies a length of less than $3\frac{1}{4}$ in. of the shaft, but contains no less than 158 constricting surfaces. The actual leakage through these glands is less than 1 per cent. of the maximum output of the turbine but this is not wasted, as a separate pipe leads it to the feed-water heater where it is economically utilised.

L M, No. 3, shows the overload bye-pass. The overload valve

introduces high pressure steam direct into the outer blade rings through a series of 60 holes drilled circumferentially in the supporting disc.

No. 8.

No. 9.

No. 10.

No. 9 shows a steam chest with the stationary dummy or labyrinthic disc attached thereto. The steam-inlet valve, overload valve, and steam outlet from the shaft packing is also shown.

No. 10 illustrates the complete series of blades, with corresponding revolving labyrinth.

A cross-section of one of the shaft bearings of the turbine is

shown in No. 11, and attention might be drawn to the delicate methods of adjustment possible by the construction of this bearing. It will be noted that there are four adjusting screws fitted with hemispheral heads. Each adjusting screw (1) is provided with a pair of locking washers (10) on which are 12 and 11 holes respectively, thus giving 132 different positions for each turn of the adjusting screw, corresponding to a thread with a very fine pitch. It will also be noted that there is an eccentric ball (22), which admits of axial adjustment by turning its corresponding screw.

No. 11.

All bearings are automatically lubricated by oil under forced pressure, and a feature of the lubricating system of the Ljungström turbine, which is common to each installation, is that it is impossible to start the turbine unless the bearings are lubricated at their normal oil pressure, as should for any reason the oil supply fail, the turbine will automatically shut down. A cylindrical chamber, by the side of the turbine, which contains the throttle valve, is shown in No. 2, and the details of same are shown more clearly in the cross-section, No. 12.

The oil pump is of the ordinary geared type, driven directly by the governor spindle, maintaining a pressure of 35 lbs. per sq. in., lowered to about 15 lbs. per sq. in. for the bearing lubrication by means of a reducing valve. The throttle valve is normally pressed

to its seat by means of a powerful spiral spring acting vertically downward. On the lower side of the valve spindle is attached a double-beat piston. The oil pressure acts on the bottom side of this piston, and consequently has to work against the spiral spring in

No. 12.

order to open the throttle valve. Thus any variations in oil pressure varies the position of the throttle valve which, through a floating lever and side shaft, maintains a constant relationship between the position of the stop valve and that of the governor, No. 13, which latter is of the Chorlton Medlock type.

An emergency release device is supplied by a steel cable under tension which, so long as it is in tension, holds shut an oil relief-valve

in the relay controlling the stop-valve; in the event of the turbine attaining an excessive speed or the bearings becoming unduly heated, the cable slackens, thus releasing a lever at the side of the stop-valve, this releases the oil pressure, the tension spring operates, and the turbine stops.

No. 13.

To start the turbine, the handle shown at the top is unscrewed a couple of turns, so as to admit sufficient steam to raise the disc valve from its seat, and allowing a sufficient amount of steam to leak past the double-beat piston-valve, which is quite sufficient to warm up the turbine in a few minutes. A few strokes of the hand pump raises the oil pressure sufficiently to lift the piston on the valve spindle against the downward pressure of its spring, so that the turbine starts, and when once it has attained its maximum speed the regulation thereafter becomes automatic.

The generators are of very substantial construction, the whole machine forming a cylindrical structure, which is adequately supported on the condenser inlet, thus forming a direct path without the

No. 14.

necessity of intermediate joints. As the turbine casing is itself part of the condenser chamber, it requires no external lagging.

No. 14 shows a cross-section of one of the generators in relation to the turbine. A ventilating fan attached to the rotor (27) draws in air from the inlet (26), the compressed air passes through slots in

the stator (22) and through the air gap between stator and rotor (24) to the outlet (29), where it can be conveniently led through ducts to the forced draught pre-heater in the boiler, dispensing with a separate fan engine. No. 15 illustrates the 1500 B.H.P. turbine, the steam chests being removed, coupled to one rotor.

No. 15.

No. 16.

The relative lightness of the rotating parts of the Ljungström turbine, coupled with their mechanical construction, should dispel any criticism respecting the overhang. In the case of the 1500 B.H.P. machine, the weight of each revolving part, that is, turbine rings, supporting disc, and attached labyrinthic packing, are only 265 lbs. and 303 lbs. respectively, while the maximum overhang to the outside of the bearing is 10½ in.

No. 16 shows the 1500 B.H.P. turbine complete with its steam chests being lifted out of the casing, and separated from its generator

No. 17.

rotors. The total weight of turbine and steam chests is about 1000 lbs. The complete turbine can be removed for inspection,

and opened up in two hours. The over-all diameter is 29 in., and over-all length across the steam chests $27\frac{1}{2}$ ins.

No insulating jackets need be removed to make bolts and nuts accessible. The internal parts of the turbine are made accessible by lifting off the upper part of the turbine casing. The couplings between the generator shafts and the turbine wheels are then unscrewed, and a clamp is applied to the steam chests. By the aid of this clamp the whole turbine and its steam chests are lifted together.

No. 17 shows a longitudinal half cross-section of the largest Ljungström turbine actually completed. Maximum output 10000 B.H.P., and speed 3000 revs. per minute. The diameter of the outer blade-ring is 34 in. The width of the outer rings necessary

No. 18. No. 19.

to give the required blade area is here obtained by building up a series of six independent blade rings, thus giving a sufficient stiffness to the drum as a whole. Each ring is connected with its neighbour through an expansion ring dovetailed and rolled together.

Nos. 18 and 19 are illustrations of the outer blade-rings of the 1500 B.H.P. and 10000 B.H.P. machines respectively, and the rigidity of the structure is clearly apparent. The cylindrical stress caused by the lineal velocity in the 10000 B.H.P. size is moderate, and the factor of safety is ample. Nos. 19 and 20 illustrate the complete 10000 B.H.P. turbo-alternator and condenser. The over-all length is 24 ft., height from bottom floor, including condenser, 21 ft., and weight of turbo-alternator 45 tons.

The overhang of each generator frame is supported by a pair of steel ties, each connected to the foundation through a stiff spiral spring or buffer, which effectively prevents any tendency to hogging

in the event of any difference of temperature between any portion of the casing, No. 4.

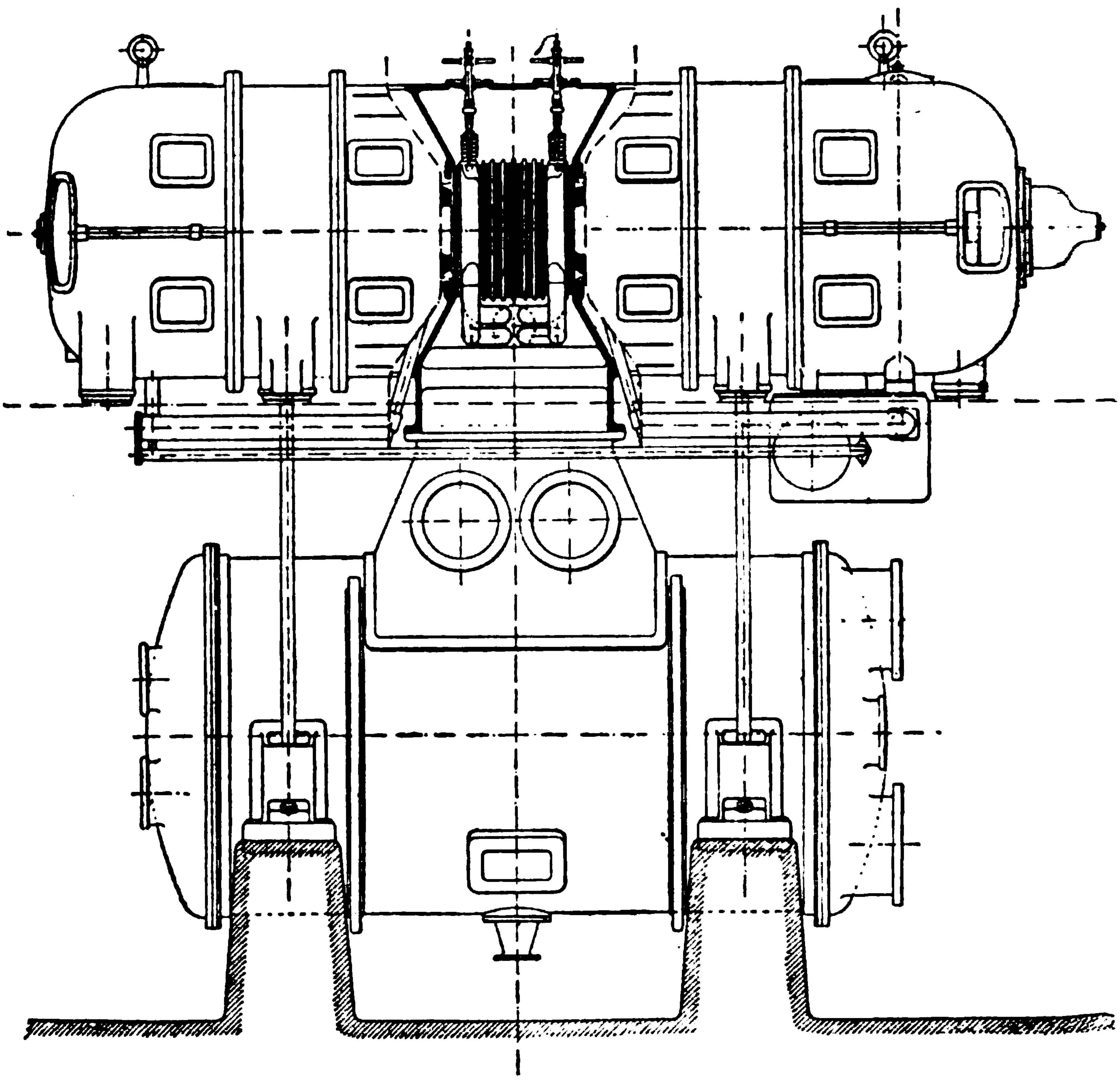
It is not possible in the limits of a paper of this nature to more than indicate the principal features of the turbine, but my endeavour has been to show the symmetry and compactness of the design and the care taken to eliminate any traces of distortion due to temperature

No. 20.

changes, so that the highest possible temperature of superheat can be used. The fact that the steam velocity is at its lowest when the temperature is greatest is in marked contrast to the ordinary type of turbine, either of the impulse or impulse reaction type, where maximum velocities and maximum temperatures are invariably associated.

The Ljungström turbine has been described as "clockwork" by critics who have seen the published drawings, but have not had an opportunity of examining the actual machine. It is true the com-

ponent parts, and consequently the weights of material, are much lighter than in any other type of turbine, but here the simile to "clockwork" ends. "Clockwork" implies complex mechanism, which is in direct contrast to the details of the design. "A harmony in engineering construction without a discordant note" was the definition of a well-known engineer. The whole turbine has been



No. 21.

so designed as to be manufactured in ordinary lathes, such as may be found in any modern engineering shop. The fitter is almost entirely replaced by the turner. The manufacture of the blade rings is ordinary repetition work, to which unskilled boy or girl labour can be trained in a few weeks. The completed machine has a factor of safety larger than in any other type of turbine. Blade stripping is unknown, due to the mechanical construction of the rings, the blades being firmly held at both ends.

SOME STEAM TRIALS OF LJUNGSTRÖM TURBO-ALTERNATORS.

Date of Test.	Normal Maximum Output in K.W.	Trial Output in K.W.	Steam per K.W.-Hour Lbs.	Steam per B.H.P.-Hour Lbs.	Pressure on Turbine Inlet (Absolute) Lbs. per Sq. In.	Superheat Deg. F.	Vacuum Per Cent.	Thermo-Dynamic Efficiency Compared with Ideal Engine. Per Cent.
Dec. 1911 -	1000	1000·5	11·6	8·3	168	302	96	76
July 1914 -	1000	990	12·7	9·0	160	150	94	77
Oct. 1915 -	6000	2000	11·1	7·9	170	270	97	79
Jan. 1916 -	3000	2700	11·15	8·0	160	280	88	87

Attention is called to the remarkably good results shown by the 3000 K.W. turbine at 90 per cent. full load. Reduced to marine conditions of 220 lbs. boiler-gauge pressure per sq. in., vacuum 95 per cent., and 240° F. superheat on the inlet valve, the steam consumption of the turbine would be 7·25 lbs. per B.H.P. per hour, or 8 lbs. per S.H.P. per hour, including generator and motor losses, but excluding condensing and propelling auxiliaries. With electrically driven auxiliaries, and including excess steam tapped from the turbine at 30 lbs. pressure for feed heating purposes, this would represent a total propelling steam consumption of 9 lbs. per S.H.P. per hour, equivalent with coal of 14000 B.T.U.'s, and an actual evaporation of 10 lbs. of steam per lb. of coal with 210° F. feed temperature to $\frac{9}{10}$ of a lb. of coal per S.H.P.

It might be said that trial conditions of consumptions may not be maintained in practice, due to possible wear of the labyrinthic edges which vary from ·004 in. to ·01 in. in the different radii of the blade rings. It has been found, however, that there is no perceptible increase in consumption. Recent trials carried out with the 1000 K.W. set installed in 1913, at the North Metropolitan Power Station, Willesden, show no increase in the original consumptions after a period of two years' continuous operation for eighteen hours a day.

In dealing with any system of marine propulsion, as indeed with any question relating to engineering, the commercial or net efficiency is the ultimate desideratum, the net efficiency in marine propulsion necessitates a consideration of the following items :—

(1) Fuel consumption.
(2) Machinery weight and space.
(3) Capital cost.
(4) Maintenance.
(5) Reliability.

Taking these factors in the order enumerated, and referring to Tables I. and II. showing the comparison of the "Mjölner" and the "Mimer," it will be seen—

(1) That there is a relative trial fuel economy of 42 per cent., and fuel economy in service of over 38 per cent., in favour of the former.

(2) The machinery weight and space shows a relative reduction of about 25 per cent.

(3) As regards capital cost, it is a sound principle that any increased cost of fuel-saving devices should be repaid in a period of not more than three years out of the increased revenue resulting therefrom, and if the propelling machinery in the electrical ship is more expensive for smaller powers than in the equivalent reciprocating ship, it will be found that the diminished fuel consumption, weight of propelling machinery, and resultant increased cargo-carrying capacity, should in normal times repay the additional cost of the installation in a period of under two years.

(4) In respect to maintenance, the experience of sixteen months running of the "Mjölner" has proved this to be extremely low. The cost of upkeep of the turbines, motors, and gearing has been practically *nil.*

(5) Reliability. During the period under review, the ship has not required to stop through any defect of the main propelling machinery, and though criticism may be raised that the vessel has not been in service for a sufficient length of time to thoroughly establish the question of reliability, on the other hand, it will be realised that this is a pioneer installation, and the entire absence of breakdown must be considered a satisfactory feature.

It is not unnatural that the principle of electrical marine propulsion should be criticised on the ground of converting mechanical energy into electrical energy at a loss of about 5 per cent., then reconverting such electrical energy into mechanical energy at an additional loss of 5 per cent. It will, however, be agreed that it is the net result which has to be considered. As reversing is effected on the motors, an astern turbine is dispensed with. The prime mover runs under the best possible thermal conditions, which include as high a superheat as can be commercially obtained without unduly sacrificing the efficiency of the boilers. The gain in thermal efficiency due to such increased temperature eliminates the electrical conversion losses. Further, the Ljungström electrical method of propulsion, in conjunction with mechanically geared motors, is a system which embodies the maximum degree of reduction between prime mover and propeller shaft, whilst the total weight of engine room machinery is lower in the smaller powers and not greater, even in the larger powered merchant ships, than any form of mechanical reduction with the ordinary impulse or impulse-reaction turbine. The reduction in weight of boiler-room machinery is obviously in direct proportion to the steam consumption of the propelling plant.

Reference is made to merchant ships in this category as in the case of torpedo-boat destroyers, or express Channel steamers, where the propeller speed is relatively high and the total S.H.P. relatively

large in comparison with the displacement of the vessel, mechanical or hydraulic reduction in some form or other still offers advantages over any system of electrical propulsion. As the field, however, which can be profitably covered by electrical propulsion ranges from merchant ships as small as the "Mjölner" to as large as the "Mauretania," it is obvious that there is a sufficient scope for the system under review.

Further, in the case of battleships or battle cruisers of sufficiently large power, turbo-electric propulsion has many advantages, given a prime mover of the efficiency of the Ljungström turbine. The generating plant can be effectively subdivided into four units, which ensures military efficiency. Economy in cruising is obtained by varying the number of poles on the propelling motors, the current being supplied from one of the generating sets.

I have already referred to the fact of the desirability of having at least two independent turbo-generators of equal power for each marine installation. Not only is this advisable in respect to the factor of safety through such duplications, but also in regard to the economy to be derived therefrom when running at reduced speeds. There is no necessity, however, in any class of merchant ship to increase the weight and expense of the motors by pole-changing devices. To run at half speed or about one-ninth power one of the two generator sets would run at half speed, absorbing about 25 per cent. of its normal full power. Between three-quarters and full speed, both sets are run in parallel at speeds corresponding to these two requirements, consequently the turbines run under at least as efficient conditions, and with the same elasticity, as in the case of ordinary geared turbines where the regulation is effected on the throttle or nozzles of the turbine.

The reversing switch and its resistances need only be used when manœuvring astern, the whole of the regulation from dead slow to full speed and *vice versa* can be effected on the turbine. The motor speed falls in direct proportion to the reduction of speed on the prime mover.

Modern three-phase generators or motors, with the windings completely enclosed in slots or tunnels in the core plates, and encased in mica-lined troughs, represent a type of machinery of a highly mechanical order. Many marine engineers—more particularly of the older school—are, I fear, not unnaturally apt to base their ideas of electrical machinery on the small lighting dynamos commonly used on cargo steamers. Commutator troubles due to dirt or overloading are, of course, non-existent in three-phase plant. I suggest that a visit to some modern electrical manufacturing works, and a close inspection of the constructional details of multi-phase generators and motors as manufactured to-day would assure the sceptic that the electrical portion of the machinery is at least as reliable as any other plant in the ship.

Electric motors aggregating many hundred thousand horse power

and of individual powers as large as 10,000 B.H.P., are to-day running on rolling mills, deep-level winding hoists, main colliery haulages, compressors, pumps, etc., where the conditions are far more severe than exist in marine propulsion, and the statistics of the insurance companies show that the breakdowns with electrical plant are con-

No. 22.—S.S. "Mjölner."

siderably less than with steam engines—conditions which are reflected in the low insurance premiums charged for large electric-power installations.

It may be of interest to know that 21 ships of an aggregate S.H.P. of 32000 are at present under construction on the Ljungström turbo-electrical system for British, Russian, Japanese, Swedish, Norwegian, and Danish owners. These vary from ocean-going

No. 23

vessels of 5400 S.H.P. to river steamers of 550 S.H.P. They include one light-draught paddle steamer of 1500 S.H.P., the power being transmitted from the turbo-alternator to two electric motors running at 450 revs. per minute. Each gearing, by single reduction to the corresponding paddle-wheel shaft, runs at from 35 to 40 revs. per minute.

TABLE I.—GENERAL PARTICULARS.

	"Mimer."	"Mjölner."
ENGINES	$19\frac{1}{2}$ in. × 32 in. × 52 in. 33 in.	Two 400 K.W. sets. 7200 revs. 3-phase. Two 500 B.H.P. motors. 800 revs. gearing on com[illegible] wheel to 85 revs.
BOILERS	Two S.E. External diam. Over-all length, 12 ft. 10 ft. 6 in.	Two S.E. External diam. Over-all [illegible] 10 ft. 10 in. 10 ft. 9[illegible]
Boiler pressure	180 lbs.	220 lbs.
Total heating surface	3150 sq. ft.	2140 sq. ft.
Grate area	83 sq. ft.	56 sq. ft.
Draught	Natural	Howden's
I.H.P. maximum at sea	1000	1140
S.H.P.	865	1000
Revolutions (propeller)	90	90
Weights—propelling machinery	Engine room and tunnel - 76 Boiler room - 100 Total - 186	55 81 136
	TRIAL RESULTS.	
I.H.P.	849	981
S.H.P.	730	843
Revolutions	86	89
Displacement	1698	1648
Mean draught	11 ft. 9 in.	11 ft. $5\frac{1}{2}$ in.
Speed (knots)	11·3	11·7
Boiler pressure	180 lbs.	220 lbs.
Draught	Natural	Howden's $\frac{5}{8}$ in. in ashpit
Superheat	Nil	190° F.
Feed temperature	—	230° F.
Vacuum (bar 30 in.)	$26\frac{1}{2}$ in.	29 in.
	COAL CONSUMPTION.	
Per hour	1313 lbs.	873 lbs.
Per day of twenty-four hours	13·1 tons	9·4
Per I.H.P. per hour	1·545 lbs.	0·890 lbs.
Per S.H.P. per hour	1·798 lbs.	1·036 lbs.
Calorific value of coal	13498 B.T.U.'s	13485 B.T.U.'s
Moisture, per cent.	4·1	2·2
Ash, per cent.	3·6	5·0

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APPENDIX

A brief description is here given of the Ljungström turbo-electrical propelling plant fitted in the "Mjölner," No. 22, and particulars are furnished of the comparative trials of the "Mjölner" with her sister ship "Mimer," the latter possessing reciprocating engines.

These two sister ships, each of about 2250 tons displacement, were ordered in 1913 for coasting trade by the Stockholm Rederiaktiebolag Svea, and delivered to their owners at the end of 1914.

The following guarantee was given:—

On trial, the consumption per I.H.P. (equivalent) per hour of the electrically-propelled vessel should not exceed 70 per cent. of the consumption per I.H.P. per hour of the standard triple-expansion engine of the other vessel under equal conditions, excepting that the turbo-electrically driven vessel should have the benefit of the reduction in weight of machinery as regards draught.

The actual trial consumption showed that the "Mjölner" only consumed 58 per cent. of the fuel of her sister ship.

During a period of six months' service conditions, the owners have certified that the coal per ton mile consumed by the "Mjölner" was 62 per cent. of the coal consumed by her sister ship for the same period.

The principal dimensions of the two boats are:—

Length between perpendiculars	225 ft.
Breadth extreme	36 ft.
Depth moulded	15 ft. 6 in.
Draught loaded	14 ft. 9 in.
Displacement	2250 tons.
Block coeff. at 14 ft. 9 in. draught	0·665
Gross tonnage	976
Net tonnage	376

Table I. shows the trial results.

Table II. shows the results of voyages with inferior coal.

There are two turbo-generator sets in the "Mjölner," each set consisting of a standard Ljungström type double-rotation turbine attached to two independent three-phase alternating current generators, one on each side of the turbine.

No. 23 illustrates one of the generating sets, with the upper turbine casing removed; exposing the turbine and steam chests and the ends of the rotors.

A small direct-current generator for supplying current to the magnets of the rotors is direct-coupled to one end of the shaft. The whole is mounted in a cylindrical casing placed direct on top of the condenser. Any leakage from the labyrinthic shaft-packings of the turbine is taken to the hot well through a water-spray feed-heater.

The turbo-generators run at a speed of 7200 revs. per minute, and supply three-phase alternating current of 500 volts at 120

alternators per second. The maximum load is approximately 400 K.W. per generator unit—*i.e.*, 800 K.W. for the complete installation. The over-all length of the turbo-alternator is 10 ft., the external turbine blade ring 15 in., and the weight of the turbo-alternator 7 tons. The length of the engine room is 20 ft.

The generator speed is regulated by centrifugal governors which act direct on the main steam supply through a throttle valve operated

No. 24.

by oil pressure. In common with all plants under the Ljungström system, the normal pressure of the forced lubrication is 35 lbs. per sq. in. on the delivery side of the oil pumps. Oil at this pressure is supplied both to the main stop valve and to the various bearings, and the turbines can neither start nor operate unless such oil pressure is normal, thus ensuring effective and complete lubrication whilst the plant is in operation.

The generating plant has been designed so as to avoid any possible trouble from salt or dust deposited on the generator stator

or rotor. The generator rotor contains no air ducts. The stator air ducts are straight, and consequently easily swept without removing or dismantling any other portion of the plant. The rotor is adequately cooled by the draught through the air-gap, such draught

No. 25

being efficiently provided by circulating the air from a fan fixed on each rotor shaft.

The ventilating air economically absorbs the heat from the generator by taking the same direct to the furnaces, by ducts leading to the Howden preheater fitted in the smokebox in the usual manner, thus dispensing with a fan engine. Mica only is used for the insula-

tion of the rotor windings, which are consequently able to withstand a considerable amount of overload without injury.

The condensers of a modified contraflo type easily maintain a vacuum of 97 per cent. in service with a sea temperature of 65° F. This high vacuum is in part due to the efficiency of the air-pump plant, and in part to the care which has been exercised in obtaining an air-tight condenser, to which the design of the turbine admirably lends itself. No auxiliary engine exhausts are led into the main condensers, and where outlets are unavoidable water locks have been fitted to exclude the entry of outside air. An electrically-driven vertical centrifugal pump draws the condensate into the kinetic tank. The condensate from the feed heaters is led to the main condensers through additional small heaters in series with the main feed-pipe.

The circulating pumps are three in number, each electrically driven by direct-coupled motors running at a speed of 3600 revs. per min. The current is supplied from the main generators.

The air pump system is of the kinetic type.

One electrically-driven rotary multi-stage feed pump is supplied for each generator unit. Each feed pump is capable of feeding both boilers under full power. The pumps are of the multiple-centrifugal type, the motor running at a speed of 7200 revs. per min., the current being also supplied direct from the main generator. The power taken by the feed pump is less than 1 per cent. of the total shaft horse power.

The pumps deliver the water to the boilers through feed heaters of the inverted condenser type. The excess steam used for feed heating is taken direct from the turbine at a stage where the pressure has dropped to about 30 lbs. above atmospheric pressure, and is thus at a point where it has delivered the greater part of its energy to the turbine.

The propelling motors are of the standard three-phase induction type with slip-springs, and run at 120 periods, 900 revs. per min., and 500 volts. The motors are connected with the driving pinions by means of slipping clutches of a novel design, which are constructed to prevent breakage of propeller blades or propeller shaft in the event of the propeller fouling block ice or any similar obstacle.

The clutches are so designed that the torque necessary for slipping is independent of the friction between the gripping surfaces, This has the advantage of preventing the clutch becoming inoperative on account of possible corrosion, due to the ship having been laid up for a lengthy period, or not being in use for any other reason.

The gearing is of the helical type. The pinions are of chrome nickel steel, and the wheel of forged carbon steel. The shafts carrying the pinions run in fixed bearings of the same type as the rotor bearings; they are designed for minute adjustment, whilst at the same time possessing great rigidity. No. 24 shows the two pinions, wheel, and one-half of the gear case. The gear case is fitted to a box section bedplate on which the two motors are supported.

Owing to the motors and gearing (No. 25), being on a common bedplate, the plant as a whole is self-contained, and all adjustments

can be made at the works of the manufacturer, thus eliminating any extra work on board ship. The lubrication of the gearing is effected by a separate oil-pump delivering the oil to the bearings and to the teeth under pressure. Means have been provided for cooling the oil, but practice has shown this to be unnecessary, as the gearing runs perfectly cool without utilising any special device. The speed of the teeth is low, about 27 ft. per sec.

A Michel thrust bearing is mounted on the forward side of the gear casing. A thrust bearing of the usual horseshoe type is fitted in the recess, but this is normally out of action, and intended only as a stand-by. It will be dispensed with in future ships.

Manœuvring is effected by the hand wheel shown on the forward side of the rectangular pillar-shaped switch-box illustrated in No. 26. If the hand wheel is moved from the central stop or mid position in one direction, the propeller turns in a similar direction, its speed corresponding to the angle through which the wheel has been turned. About 120° represents full power ahead, and 120° in the other direction represents full power astern.

The regulations of the speed of motors when reversing are effected by inserting the rotors of the motors in series with resistances of the liquid type. These resistances consist of two sets of cones, of which the lower ones are stationary and filled with a saline solution, whilst the upper cones are mounted on vertical spindles which can be raised or lowered into the lower cones by the hand wheel, thereby inserting a greater or lesser resistance in the circuit. In mid or stop position, the upper cones are lifted entirely out of the liquid and lower cones, thereby the current is broken and the motors come to rest. At full speed the two cones come nearly into contact, the resistances being simultaneously cut out through a short circuiting switch.

Reversal is effected with the wheel in mid or stop position—that is, when the rotor is open circuited, and is simply accomplished by the change-over switch which reverses the phases in the main or stator current. A saline solution of dilute potash is used as the liquid resistance. This solution is circulated and cooled by a small centrifugal pump which is mounted on the same spindle as the circulating pump. The cones are of nickel and thus non-corrosive, and are counterbalanced, so that the power required to turn the hand wheel is nominal.

The switches are of the ordinary open type, the tension in the rotor current of the generator being automatically lower before the circuit is broken; they are found to work perfectly satisfactorily. For larger purposes, oil switches are used.

The elasticity of control is considerable, the speed of the propeller ranging from about 12 revs. per min. to 90 at full speed. The total time of propeller reversal from full speed ahead to full speed astern was carefully tested on the trial trip of the "Mjölner," and averaged fifteen seconds.

A general illustration of the engine room, showing the starboard turbo-generator, motors, and gearing, is shown in No. 27.

No. 20.

Drawing No. 28 is for a twin-screw vessel of about 21000 tons displacement and 13 knots sea-speed, where an equivalent of 6250 I.H.P. is required. The machinery consists of two Ljungström

partments aft and remote from the turbo-alternator and boiler compartments. The latter compartments are placed amidships, two being allotted to the turbo-alternators with their auxiliaries, two to the boilers, and two to the bunkers. Between the two latter independent groups are placed cargo holds of any desired length,

with suitable water-tight bulkheads. The cables between the alternators and motors are carried along the deck and about the middle line of the vessel. These cables are independent for each motor, but are cross-connected at a suitable position.

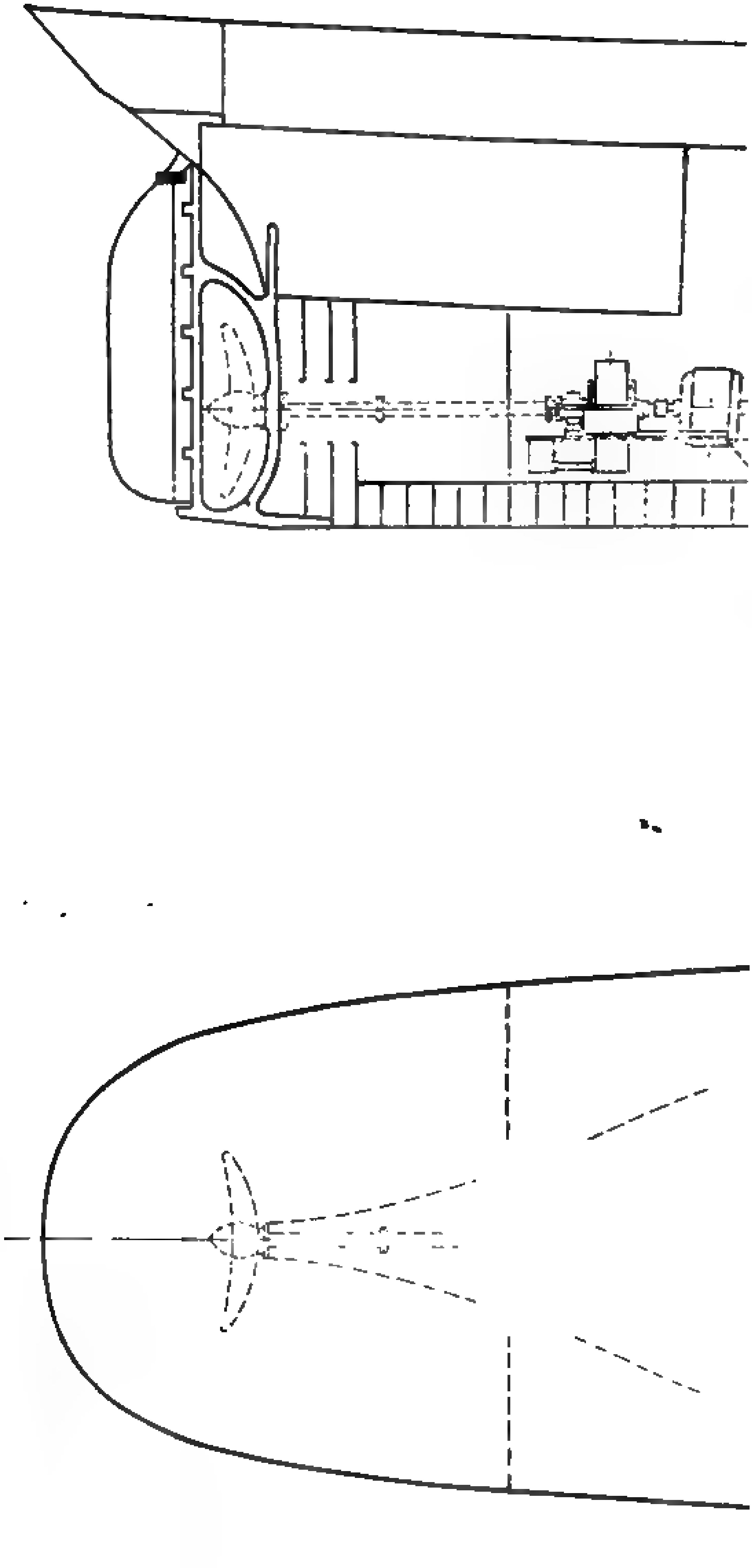
The boilers are of the cylindrical type arranged for Howden's system of forced draught and for burning coal as fuel. The heating surface is 13000 sq. ft., boiler pressure 220 lbs., and total steam temperature 630° F. The condensing plant is designed for a 95 per cent. vacuum. The whole of the above machinery weighs 830 tons "steam up," and is capable of developing power equivalent to 6250 I.H.P. on a consumption of 60 tons per day of Welsh coal.

The flexibility of the turbo-electric system obviously lends itself to the most efficient sub-division of the ship into water-tight compartments. Thus, if one motor room, one boiler room with bunker, and one engine room were flooded, the ship could still run at about 70 to 75 per cent. of her normal speed with the remaining machinery. In addition to the water-tight sub-division, a further great advantage is the absence of shaft tunnel and the consequent increase in cubic contents of the after-holds, together with greater ease in handling cargo due to the absence of these tunnels.

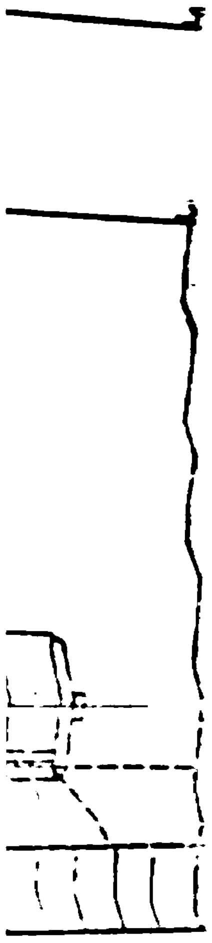
The following table illustrates the most important features of the system as compared with reciprocating engines, cylindrical boilers and superheaters being used for the turbo-electric system, and cylindrical boilers with saturated steam used for the reciprocators.

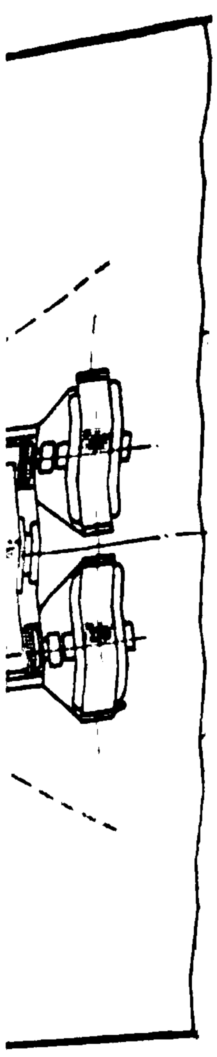
Item.	Ljungström Turbo-Electric System.	Reciprocating Engines.
Coal per day (tons) - -	64	96
Coal per I.H.P. hour (equiv.) -	0·96 lbs.	1·43 lbs.
Total heating surface - -	13000 sq. ft.	18000 sq. ft.
Total grate area - - -	300 sq. ft.	426 sq. ft.
Weight of machinery (tons) -	830	1·400
Weight of thirty days' bunkers (tons)	1920	2880
Gain in dead weight (tons) -	1530	...

The second drawing, No. 29 illustrates the machinery for a cargo steamer of 6500-7000 tons dead weight, in which 1750 equivalent I.H.P. is required to drive the ship at about 10 knots. In this case the boilers are placed in a separate water-tight compartment on the middle line of the ship, with the two three-phase turbo-alternators on deck above the boilers and exhausting to the condensers, which are situated on the tank top level and on either side of the boiler room. By this arrangement a considerable amount of cargo space is saved, and it also permits of the machinery being operated even in the extreme case where both condenser rooms are open to the sea, as the turbines will automatically exhaust to atmosphere, and the



No. 29.—Arrangement of Machin





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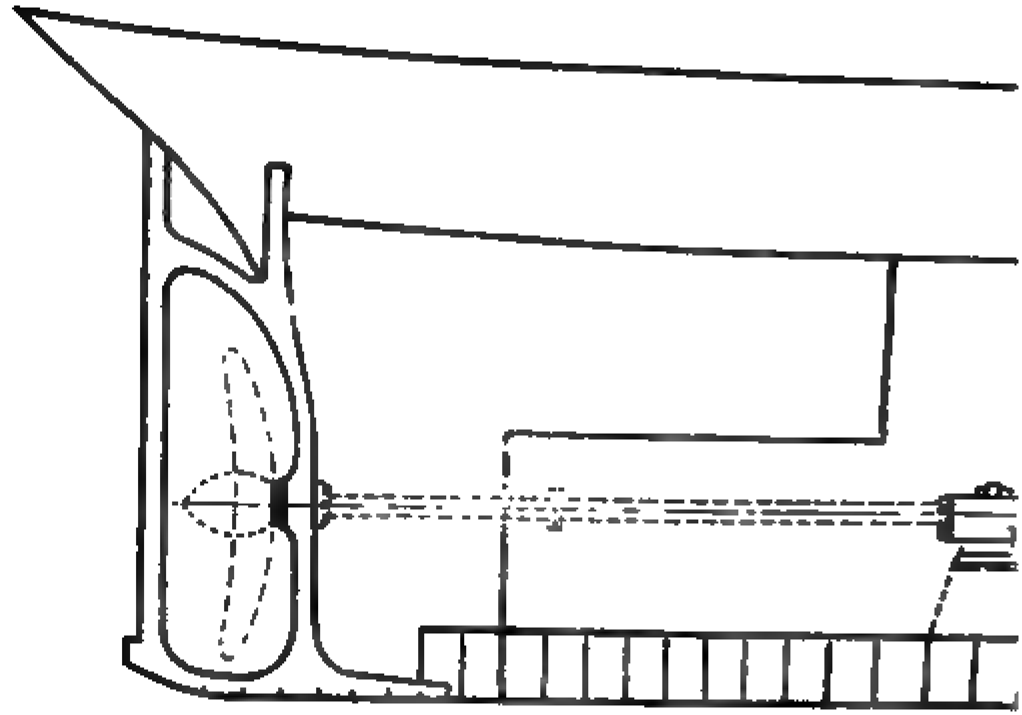
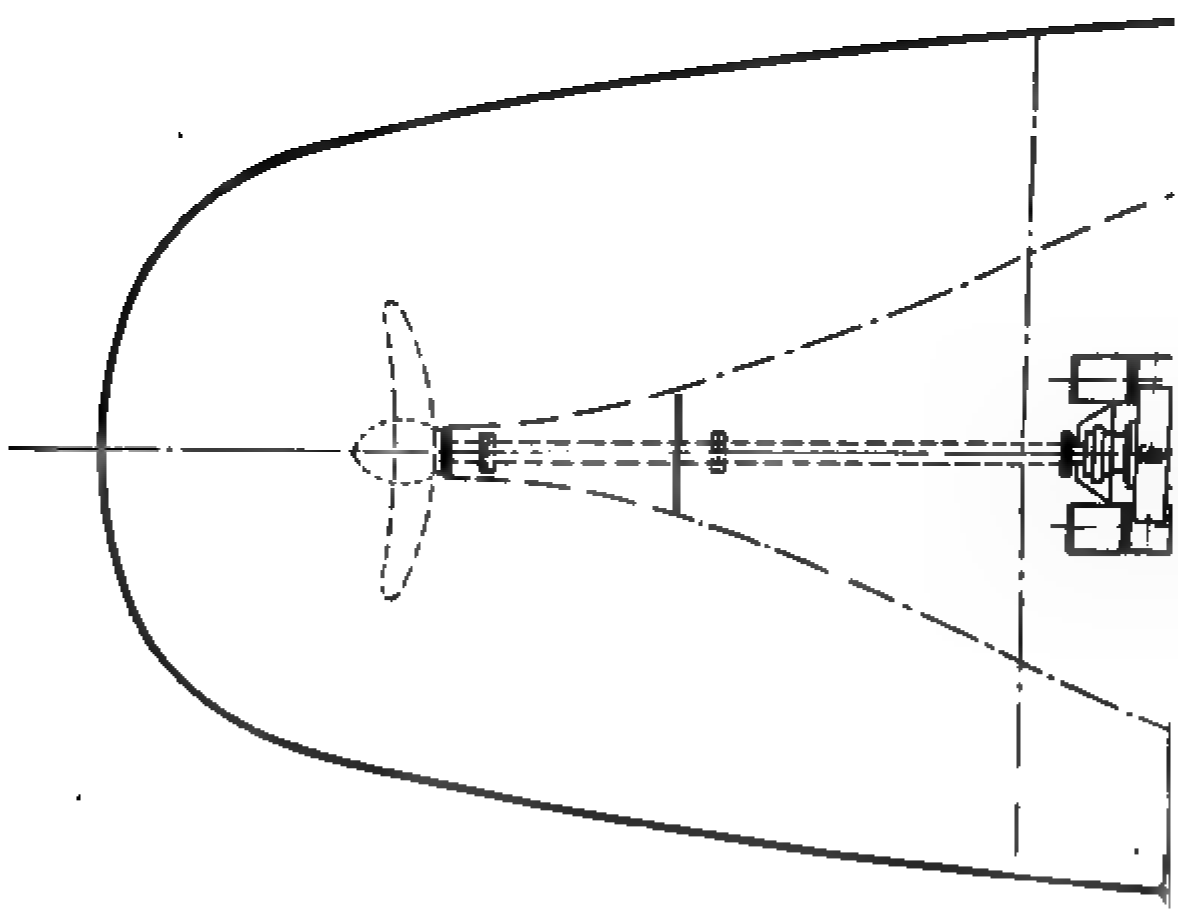




No. 30.—Proposed Arrangement

› can still be driven at reduced speed. The propelling motors geared to the shaft through double helical single reduction gear, l are placed in a separate compartment as far aft as possible so to give the after-holds greater capacity and allow cargo to be ıier worked by the absence of the shaft tunnel. The following ›le shows a comparison between Ljungström turbo-electric system d reciprocating engines of the standard type:—

	Ljungström Turbo-Electric Scheme.	Reciprocating Engine, Standard Type.
I.H.P. } At sea	1750 (equiv.)	1750
S.H.P. } At sea	1575	1575 (equiv.)
R.P.M. propeller	76	76
Weight of machinery - -	195 tons	345 tons
Coal per day (Welsh) - -	18 tons	28 tons
Boiler pressure - - -	220 lbs.	185 lbs.
Superheat - - - - -	230° Fahr.	Nil.
Vacuum - - - -	28½ in.	26 in.
Steam per I.H.P. hour - -	9 lbs.	15 lbs.
,, S.H.P. hour - -	10 lbs.	16·6 lbs.
Coal per I.H.P. hour - -	0·95 lbs.	1·5 lbs.
,, S.H.P. hour - -	1·07 lbs.	1·66 lbs.
Heating surface - - -	3600 sq. ft.	5350 sq. ft.
Grate area - - - -	85 sq. ft.	130 sq. ft.
Draught - - -	Howden	Howden
Ratio - - - - - -	1 : 42½	1 : 41
Coal per square foot grate -	20 lbs.	20 lbs.

Drawing No. 30 shows the machinery arranged in a manner similar to that in drawing No. 29, but in this case the boilers are arranged for burning oil in the furnaces, and motor-driven pumps are indicated for pumping the tanks should the ship be designed to carry oil in bulk. The total propelling power at sea in this instance is 2800 equivalent I.H.P. Apart from the direct advantage of the turbo-electric propulsion in this case, further gain is obtained by pumping the cargo oil by means of electrically driven pumps. No extra generating plant is required, as one of the main propelling sets may be used for this purpose, running at one-third load, in which case the consumption is 14 lbs. per K.W. hour. The nett result would be that the ship will be in port about 33 per cent. less time, and discharge about 60 per cent. more oil per hour, at a total cost of about one-half as compared with the ordinary steam-driven pumping system adopted in oil tank steamers.

In the three designs illustrated by the drawings the propelling auxiliaries, steering gear, and deck machinery are electrically operated, and may be supplied with current from either of the main sets through static transformers or from the auxiliary sets indicated.

Manœuvring is entirely accomplished on the stop valve of the turbine, the speed of the propelling motors following speed of turbine in exact ratio.

Reversing is effected by reducing speed of turbo-alternators about 20 per cent., and then throwing over reversing switch on motors by means of a relay connected to stop valve. No intermediate resistance is required, and the motors are of the squirrel cage type.

Table of Comparative Steam Consumption for Vessels of 5400 S.H.P. Normal Sea Power. (*Compiled by R. S. Portham, Esq.*)

(1) = Ljungström turbo-electric drive, 630° Fahr. turbine inlet, 220 lbs. boiler pressure, 95 per cent. vacuum, forced draught.

(2) = Mechanical-geared impulse or disc-drum turbines, 460° Fahr. turbine inlet, 200 lbs. boiler pressure, 95 per cent. vacuum, forced draught.

Twin screw, 13 knots at sea, 21000 tons displacement, propeller revolutions per minute 90.

(1).	Lbs. of Steam per Hour.	(2).	Lbs. of Steam per Hour.
Turbo-alternators, fan, propelling motors, gearing, and thrust block = 9·25 lbs. per S.H.P. per hour	49950	Turbines, gearing, and thrust block = 10·75 lbs. per S.H.P. per hour	58100
Electrically driven.		Independent aux. engines.*	
		Fan engine	1200
Air and circulating pumps	960	Air and circulating pumps	4500
Feed pumps	450	Feed pumps	1850
Steering gear	100	Mean. Steering gear	850
Mean. Electric lighting, 75 kw.	150	Mean. Electric lighting, 75 kw.	550
Mean. Bilge, ballast, sanitary make up, and lubricating oil pumps	200	Mean. Bilge, ballast, sanitary, make up, and lubricating oil pumps	1000
Mean. Refrigerating CO_2 plant	600	Mean. Refrigerating CO_2 plant	2400
	52410		70450
Total steam tapped from turbine for heating feed from 95° Fahr. to 210° Fahr. = 11½ °/₀ of 52410, or 6030 lbs. per hour, less 40 °/₀ useful work done in turbine, or 2410 lbs. = 3620 lbs.	3620	Total steam required for feed heat = 11½ °/₀ of 70450. = 8100 lbs. per hour.	
		* Total auxiliary steam = 12350 lbs. per hour, or 34 °/₀ of total auxiliary steam lost in condensation and radiation. Auxiliaries exhaust to feed heater to raise temperature from hot well 95° Fahr. to 210° Fahr.	
Total steam, all purposes	56030	Total steam, all purposes	70450
Per S.H.P. per hour =	10·40	Per S.H.P. per hour =	13·03

Contraflo type. The two main induction motors are coupled to the propeller shaft through double helical gearing made by the Power Plant Company Limited, West Drayton, and are together capable of delivering the 1500 S.H.P. continuously at sea with a propeller speed of about 76 revs. per min. The stators are wound with single bars per slot, and insulated with seamless mica tubes, the end connections being of heavy copper strip. The rotor windings are of the cylindrical barrel type. Each motor is ventilated by its own fan, which is mounted on the rotor spider. The main bearings are of the pedestal type, with split spherical bushes, and are arranged for forced lubrication. The brush lifting and short-circuiting gear are mounted on the forward end of the motor shafts, and are operated by handle interlocked with the starting

hand-wheel. The shop tests gave an efficiency of 95 per cent. at full load, the corresponding power factor being 0·875. The turbo-alternators, condensers, main motors, auxiliary motors, and manœuvring gear were manufactured by the Brush Electrical Engineering Company Limited, Loughborough, to the specifications of the main contractors—the British Ljungström Marine Turbine Company Limited, under the supervision of Messrs Esplen.

The auxiliaries are, for the most part, electrically driven, and in this connection reference should be made to the two electrically-driven centrifugal circulating pumps, each driven by three-phase, 60 cycle motors, at 650 volts, and about 1750 revs. per min. There are two air pumps, one being of the kinetic type, with the condensate and kinetic pumps embodied in the one casing, each combination being driven by a three-phase vertical motor at about 1470 revs. per min. Both circulating and air pumps are supplied by the Pulsometer Engineering Company Limited. The electrically driven feed pump is of the turbine type, supplied by the Power Plant Company, and is capable of feeding both boilers at full power. It is driven by a three-phase motor at 3540 revs. per min. An auxiliary-driven feed pump by Clarke, Chapman & Co., intended for harbour use, can be connected to the main feed discharge to supply the boilers as a standby. The boilers are fed through a feed heater designed by the Contraflo Condenser and Kinetic Air Pump Company, and the necessary steam for this purpose is tapped from the main turbines and drained to the kinetic tank. The soda pumps for circulating the electrolyte, supplied by the Pulsometer Engineering Company, are mounted on the manœuvring gear, and are driven by alternating current motors at about 1600 revs. per min. The electric lighting set is of the combined type, in which a direct current 18 kw. generator may be driven either by a three-phase motor or by a de Laval steam turbine, the three units being mounted on one shaft with a suitable disconnecting clutch. In harbour the set is driven by the 25 B.H.P. turbine, whilst at sea the motor takes its current from the main generating set. In addition to supplying the electric lighting set and wireless plant, this set is capable of exciting both propelling turbo-alternators.

For ballast and general service purposes two duplex pumps are fitted in the engine-room, a further pump being installed for fresh water purposes. A small donkey boiler is fitted between the two main boilers for use in harbour, and is designed to work at 100 lbs. pressure with saturated steam. The deck winches are driven by steam, but should electric winches be fitted at some future date, no additional generator plant will be required, as one of the main propeller sets running at one-eighth of full load will serve, and even at this low fraction the consumption is only 18 lbs. per kilowatt hour. The control panels for the electrically-driven auxiliaries and the lighting set are situated on the after engine room bulkhead.

The auxiliary condenser is of the Contraflo non-vacuum type, capable of taking the exhaust from the deck machinery and harbour auxiliaries. It drains to a cascade filter fitted with the usual bye-pass and float control to the feed pumps, this plant being of the Richardsons-Westgarth type. A 20 ton Morison's evaporator is installed in the engine room, and is so arranged that the feed supply may be taken from one of the auxiliary pumps, or from the sanitary tank.

SECTION XVIII.

RECENT DEVELOPMENTS IN MARINE TURBINE PRACTICE.

The writer is indebted for much of the information which follows to the admirable and scholarly Presidential Address of Mr Alexander Cleghorn, delivered before the meeting of the Institution of Engineers and Shipbuilders in Scotland on 23rd October 1917, and has to thank Mr Cleghorn for most kindly granting permission to reprint largely from the paper referred to, which contains a general résumé of improvement in turbine design during the past ten years.

During the past few years considerable advance has been made in turbine practice, and this progress has resulted in still further economy in fuel and water consumption per horse-power hour. Owing to the rapidly extending use of gearing, marine turbine design is now, in many respects, similar to that of electrical practice, and the original type of reaction drum turbine is gradually falling into disuse, and giving place to that of the impulse wheel type.

This change is permissible where gearing down is fitted, as with either single or double gear down pinion gear a high speed of turbine and low speed of driving shaft can readily be arranged for.

As the reaction drum turbine was specially designed to permit of moderate shaft speeds for direct drives, it will easily be seen that if high turbine speeds are desirable the impulse type will meet the case, and give satisfactory results in every way. Hence, as may be said, the "reversion to type" which has recently taken place in marine turbine design, which, as before remarked, now approaches that of electrical turbo-generator practice.

Again, it should be noted that for a given power the faster the turbine speed the less the size of rotor required, so that with gearing down the size of turbine necessary is much reduced owing to the higher revolution speed.

Speaking generally, the combination system, and the direct turbine drive system are both obsolete, the gearing down arrangement being fitted in practically all vessels under present construction or order.

Modifications in Turbine Design.

To suit the new conditions of mechanical gearing, the Parsons type of turbine as fitted is usually of the combined impulse and reaction pattern, with one impulse wheel at the H.P. initial end, and

a drum with reaction blading for the succeeding stages. Compared with the old direct drive turbine, the L.P. turbine expansions, although increased in number, each contain less blade rows than formerly, the number per expansion ranging from two to three, and at certain positions, of one blade row only.

The following gives a fair general idea as to the expansions in a modern reaction geared down turbine set.

PASSENGER STEAMER.

H.P. Turbines (2).		L.P. Turbines (2).	
Expansion.	Blade Rows.	Expansion.	Blade Rows.
1	12	1	2
		2	2
2	12	3	2
		4	2
3	12	5	2
		6	2
4	6	7	2
		8	2
5	6	9	2
		10	1
6	6	11	1
Blade heights range from $1\frac{1}{4}$ in. to $2\frac{3}{8}$ in.		Blade heights range from $1\frac{1}{4}$ in. to 10 in.	

Turbine Drive and Turbo-Electric Drive.

The latest developments in marine steam turbine practice comprise the following:—

TYPE A. **Turbines.**

Boilers.
1. Water tube type.
2. Oil fired.
3. Superheaters (150° superheat at stop valve).

Turbines.
1. High revs. speed impulse type turbines.
2. Double reduction flexible type gearing.
3. Rotary or ejector type air pumps.

TYPE B.

Turbo Electric Drive.

Boilers.
1. Scotch or water tube type.
2. Coal or oil fired.
3. Fitted with superheater (250° superheat at stop valve).

Turbines and Generators.	1.	High speed impulse type, Or Ljungström type turbines.
	2.	Electric generators and motors.
	3.	Single reduction gearing.
	4.	Rotary or ejector type air pumps.

Comparison of Impulse Turbines for Direct Drive and for Geared Down Drive.

Direct drive turbines	1. Of large diameter and weight, running at low revolution speed. 2. Fitted with short blades. 3. Arranged with a number of moving blades at each stage wheel and a corresponding number of guide blades (one row less) in the casing.
Geared-down turbines	1. Of small diameter and weight, running at high revolution speed. 2. Fitted with longer blades. 3. Arranged with moving blades and guide blades (one row less) at first expansion stage wheel only, and leaving one row only of moving blades on each wheel of following stages. 4. The increased speed of rotation is counterbalanced (as regards breakdown) by simplicity of design and reduced number of detachable parts, also by improved running balance.

As the efficiency of a turbine, within certain limits, varies inversely as the number of blade rows or velocity stages, it therefore follows that a high speed turbine supplied with the minimum number of blade rows per stage wheel will give higher efficiency than one fitted with a greater number of moving blade rows at every stage wheel.

Characteristics of Marine Type Turbines.

(1) Parson's Reaction Turbine. — Consists of parallel drums with reaction guide and moving blades.

Mean blade speed = 120 ft. per sec.
,, steam ,, = 270 ,, ,, ,, } **Ratio** $\frac{Vt}{Vs}=\frac{120}{270}=\cdot 44.$

(2) Parson's Combined Impulse and Reaction Turbine. — Consists of one impulse wheel on H.P. and astern turbines only, the impulse wheel being velocity compounded, and the remaining expansions formed of reaction blading on parallel or stepped drums.

Mean blade speed = 170 ft. per sec.
,, steam ,, = 300 ,, ,, ,, } **Ratio** $\frac{Vt}{Vs}=\frac{170}{300}=\cdot 56.$

(3) Curtis Impulse Turbine. — Consists of a series (five to seven) of impulse wheels, each velocity compounded, the full expansion of the steam being completed in the single turbine.

Mean blade speed = 250 ft. per sec. ,, steam ,, = 1500 ,, ,, ,, } **Ratio** $\frac{Vt}{Vs} = \frac{250 \times 3}{1500} = \cdot 5.$

NOTE.—As the wheels are of the 3-stage velocity type, multiply the blade speed by 3 for combined blade speed.

(4) Brown-Curtis Turbine A. — Consists of four or five velocity compounded impulse wheels, followed by further impulse stages fitted on a drum, either parallel or stepped.

Similar to last type.

(5) Brown-Curtis Turbine B. — Consists of four or five impulse wheels, velocity compounded, followed by a large number of few blade row reaction expansions, fitted on the drum stage, the latter being stepped down. When separate H.P. and L.P. turbines are fitted, the H.P. is formed of from four to six impulse wheel stages only.

Mean blade speed = 170 ft. per sec. ,, steam ,, = 350 ,, ,, ,, } **Ratio** $\frac{Vt}{Vs} = \frac{170}{350} = \cdot 48.$

(6) Brown-Curtis Turbine C. — H.P. Consists of six or seven impulse wheels, the first velocity compounded, the others single stage (one blade row).
L.P. turbines fitted with simple impulse stage wheels only. Reverse turbine with single wheel, velocity compounded. This type is directly adapted from electric practice, and is specially suited for high revolution speeds, say 3000 to 4000 revs. per minute, such as are common with double reduction gearing.

Mean blade speed = 450 ft. per sec. ,, steam ,, = 900 ,, ,, ,, } **Ratio** $\frac{Vt}{Vs} = \frac{450}{900} = \cdot 5.$

(7) Rateau Turbine. — Similar in many respects to the Curtis, and consists of a series of single impulse stages, the first only, being, in some cases, velocity compounded.

Mean blade speed = 450 ft. per sec. ,, steam ,, = 850 ,, ,, ,, } **Ratio** $\frac{Vt}{Vs} = \frac{450}{850} = \cdot 53.$

(8) Ljungstrom Turbine (Turbo-Electric Drive). — Consists of a pair of opposed bevelled steel discs fitted with concentric rings of radial flow reaction blading. The discs revolve in opposite directions, and drive two generators, placed one at either end of the turbine. This type is by far the smallest turbine for a given power yet produced.

Mean blade speed ranges from about 80 ft. to 850 ft. per second (1500 S.H.P. size) according to radial position.

Mean steam speed ranges from about 400 ft. to 1200 ft. per second, varying with radial position.

$$\text{Then}\begin{cases}\text{Ratio } \dfrac{Vt}{Vs} = \dfrac{80}{400} = \cdot 2 \text{ near centre (at inner blade rings).}\\[2ex]\text{Ratio } \dfrac{Vt}{Vs} = \dfrac{850}{1200} = \cdot 7 \text{ near periphery (at outer blade rings).}\end{cases}$$

$$\text{Mean ratio } \frac{Vt}{Vs} = \frac{\cdot 2 + \cdot 7}{2} = \cdot 45.$$

$$\text{Mean of Means} = \frac{\cdot 2 + \cdot 7}{2} = \cdot 45.$$

Power Developed in Impulse and in Reaction Stages.

In a combined impulse and reaction turbine the impulse stage develops from $\frac{1}{5}$ to $\frac{1}{4}$ of the total power, and the reaction stages develop the remaining $\frac{4}{5}$ or $\frac{3}{4}$ power of the turbine.

Quoting from Mr Cleghorn's Paper:—

"Advances in turbines themselves have been largely due to continual small improvements in design and construction, although the step from the direct-coupled to the geared type might, in effect, be rated as a progression of fundamental distinction. At these higher speeds, the axiomatic condition of the smallest possible rotor at efficient blade speed can be readily attained.

"In both reaction and impulse types reduction in diameter means larger blades, and relatively much lower tip or diaphragm leakage. The rotors are much more rigid, and clearances may be considerably reduced, with still less prospect of distortion troubles. The applicability of superheat is, for these reasons, more manifest; and the smaller sizes render any extra cost, arising from the use of special materials, less significant.

"With the high revolutions used in double-geared turbines, it is customary to split the high-pressure end into two distinct portions, a high-pressure unit and an intermediate-pressure unit, either driving the same or separate pinions. The gain in rigidity and compactness is great, but to a large extent is necessary in order to safeguard the essential margin of safety below the critical speed.

"The reaction type, made up of these small units, lends itself to certain modifications in details which constitute the essential elements contributory to the increased economy claimed for it over most single-geared designs. The small and short rotors allow the successful application of special blade shrouding and packing, which transforms the wasteful tip leakages into more moderate axial losses. This particular form has been termed 'end-tightened blading,' and it does away with fine radial clearances by introducing close axial adjustments, which are more easily maintained.

"Again, the small size of these units results in a great reduction of the end thrust, and permits the total pressure to be carried by a minute Michell collar. This arrangement eliminates the dummy

element of the rotor, does away with the objectionable dummy leakage and removes the necessity to maintain the fine dummy clearances necessary for high economy.

"This is a distinct gain; and, in view of the relative merits of the two turbine types under ordinary construction, would seem to point to a certain superiority of the reaction form, where arranged in small high-speed units. The impulse type, while profiting from the reduction of size, does not make a definitive gain by the complete elimination of a source of loss. At the same time, it always possesses the popular advantages of having much larger radial clearances throughout, and has practically no casing blades—two features which make a powerful appeal in marine work, where safety is a dominant requirement.

"The thermodynamic design of turbines—while not of a scientific finality—is of an excellent practical completeness. The conditions of of steam flow, pressures, and consumptions are foretold with great accuracy, and anticipated powers are attained within a negligible margin.

"The actual troubles may be broadly grouped under two classes—the metallugical and the dynamical.

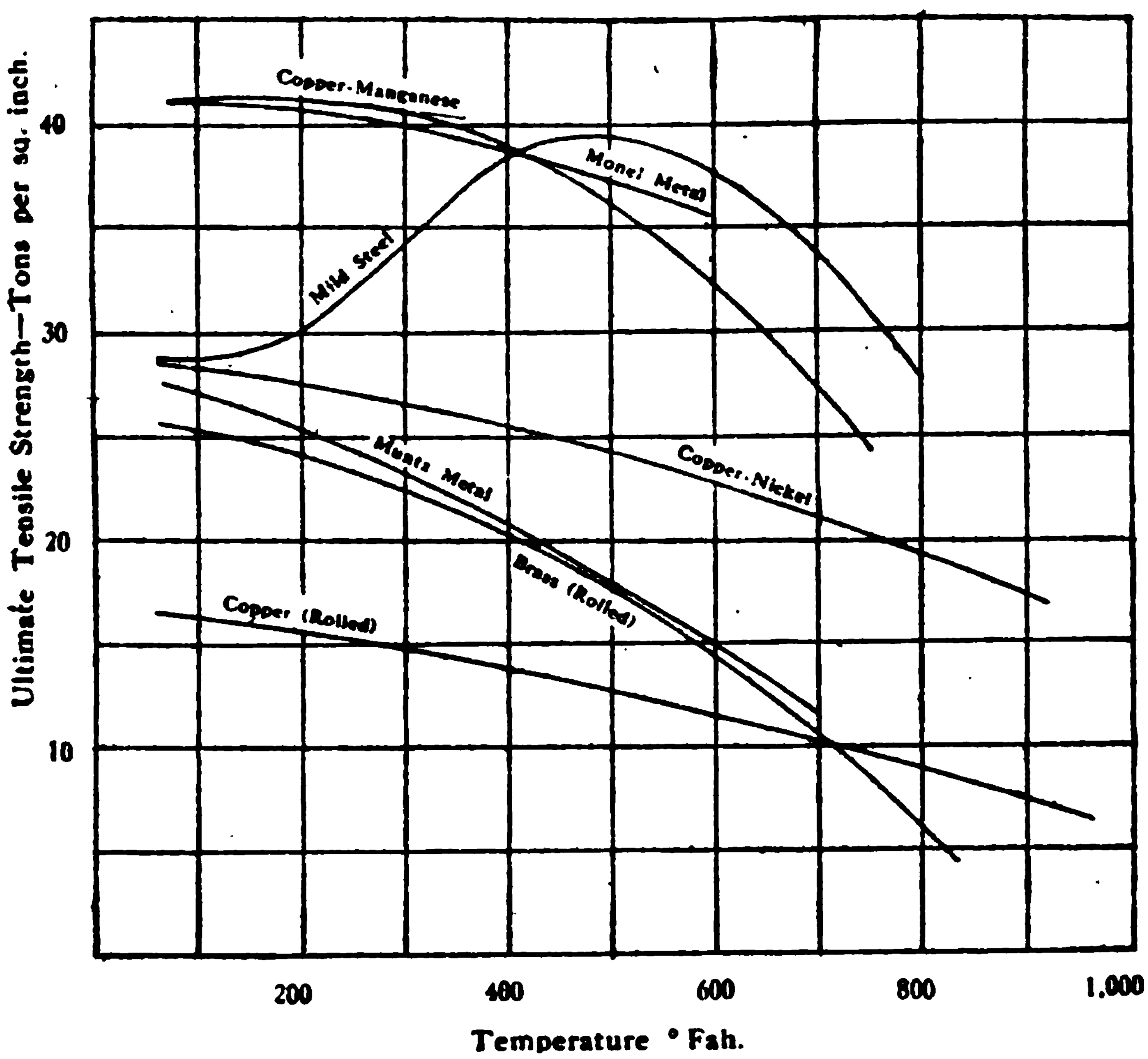
"The former, when saturated steam is used, are almost wholly confined to the blading material, and are to a large extent the outcome of growth of speeds and temperatures. The perfect blading material must be suited to withstand all the conditions of the full expansion range. Fitted at the high pressure ends it must be capable of meeting high temperatures, whether of saturated or superheated steam, and high steam speeds with their consequent weakening and 'scouring influences. In the lower-pressure sections, erosion by water must be faced; and throughout the whole range the steady and severe centrifugal stresses which are set up, repetitive bending stresses which are possible, and the indeterminate conditions of vibratory action must be allowed for.

"These demands indicate transcendent qualities of strength and hardness on the part of the material and inherent resisting power under widely varying influences, added to the special requirements for the purposes of manufacture. It seems an impossible specification to comply with, but the definite reliability and maintenance of efficiency of the modern turbine under the increasingly severe tests of high speeds and growing temperatures depend on its discovery.

"For the purposes of blading design there is a great dearth of information on the qualities of the material as modified by the working conditions. It is not sufficient to know that a substance, when pulled apart in a testing machine, records an elastic limit of 12 to 14 tons per square inch and an ultimate strength of 2 to 2½ times that amount, even if the alloy in working conditions is allowed a factor of safety of 2½ to 3 on the yield point. The elastic limit at the temperature of action must be known. At these temperatures the fatigue limit with repeated loading must be known. How does the ductility change under the conditions of working, and what is the effect of the age factor? These are all questions demanding atten-

tion, even for blading alloys now in use; and they denote the necessity for a vast amount of research additional to that of the metallurgical chemist.

"Of materials in present or recent use we have pure copper, copper-manganese, monel metal, phosphor-bronze, and 5 per cent. nickel steel. High percentage nickel steels—so common at one time—have been proved useless for blades, and the recent disclosure of experience with them in Rand power plants * is a final comment. Of the first four named, pure copper is now vanishing, and all four are



No. 2.—Curves Showing Reduction of Tensile Strength at High Temperstures.

subject to weakening with temperature and time. This may not occur to the same extent with steel, but it is very liable to rust, and probably to water erosion.

"In illustration of the temperature effects I append a diagram showing the reduction of ultimate tensile strength of several well-known materials consequent on a rise of temperature. The results embodied have been collected from various sources, and the curves

* "Deterioration of Turbine Blading," by A. Fenwick, *Engineering*, vol. ciii pp. 419, 461.

have been faired, and are therefore only approximately correct, but they clearly indicate the rapid reduction of tensile strength as high temperatures are reached. Information of a systematic character is yet lacking regarding the change of the elastic limit with temperature, and the influence which time and fatigue has in depreciating the original properties of these metals. Such information, in order to delineate the characteristics of a metal, is necessary before a rational decision can be arrived at for its adoption, and should be the subject of immediate research.

"The dynamical problem is many featured, but leads nearly always to the singular manifestation of troublesome vibrations; and hence there arises a certain difficulty in the tracing of causes when this unfortunate result occurs.

"The condition of inherent instability of the rotor, which causes the phenomenon known as 'whirling,' is well understood, and the 'critical speed' at which it arises can be closely calculated. It is usual in marine work to keep the critical speed at least 30 per cent. above the running speed. The only ambiguous point in the determination is that regarding the nature of the end supports of the rotor, and it is best, therefore, to assume that the span is the distance between the bearing centres. The critical speed in high-speed turbines exercises a controlling influence, as a design may be wholly worked out on the basis of a conjectural margin of safety in respect to the critical speed, but the whirling calculation is the final test of its suitability.

"Adventitious causes of vibration are numerous. Any lack of balance—whether original or of subsequent development—is definitely productive of trouble. The faulty alignment of the bearings, the inadequate fit of highly stressed discs on their spindle, the imperfect or neglected dynamic balance, even when the static test is excellent, large bearing clearances and inexpert coupling up to the drive are all possible constructional errors leading to vibration, should the speed be suitable. Unequal thermal expansion, non-homogeneity of material, distortion of seatings in ship, and even excessive initial strains in spindles may indicate their existence in this objectionable manner.

"It will, therefore, be seen that the causes of vibration are of a complex nature, difficult to guard against in construction, and troublesome to detect on actual occasion; and when it is recognised that this phenomena persistently shows itself in high-speed work, the necessity for painstaking research to fathom the causes and discover methods of elimination, where such are possible, will appeal to all."

TURBINE TY

Turbines ma

Type.	Rea
Construction	Drum f movi and c witl blade
Example -	Pa
Application -	Marine sion.

For direct tur
the use of superhe
design to No. 4, b
consisting merely
general practice, t
conditions favour

Turbines

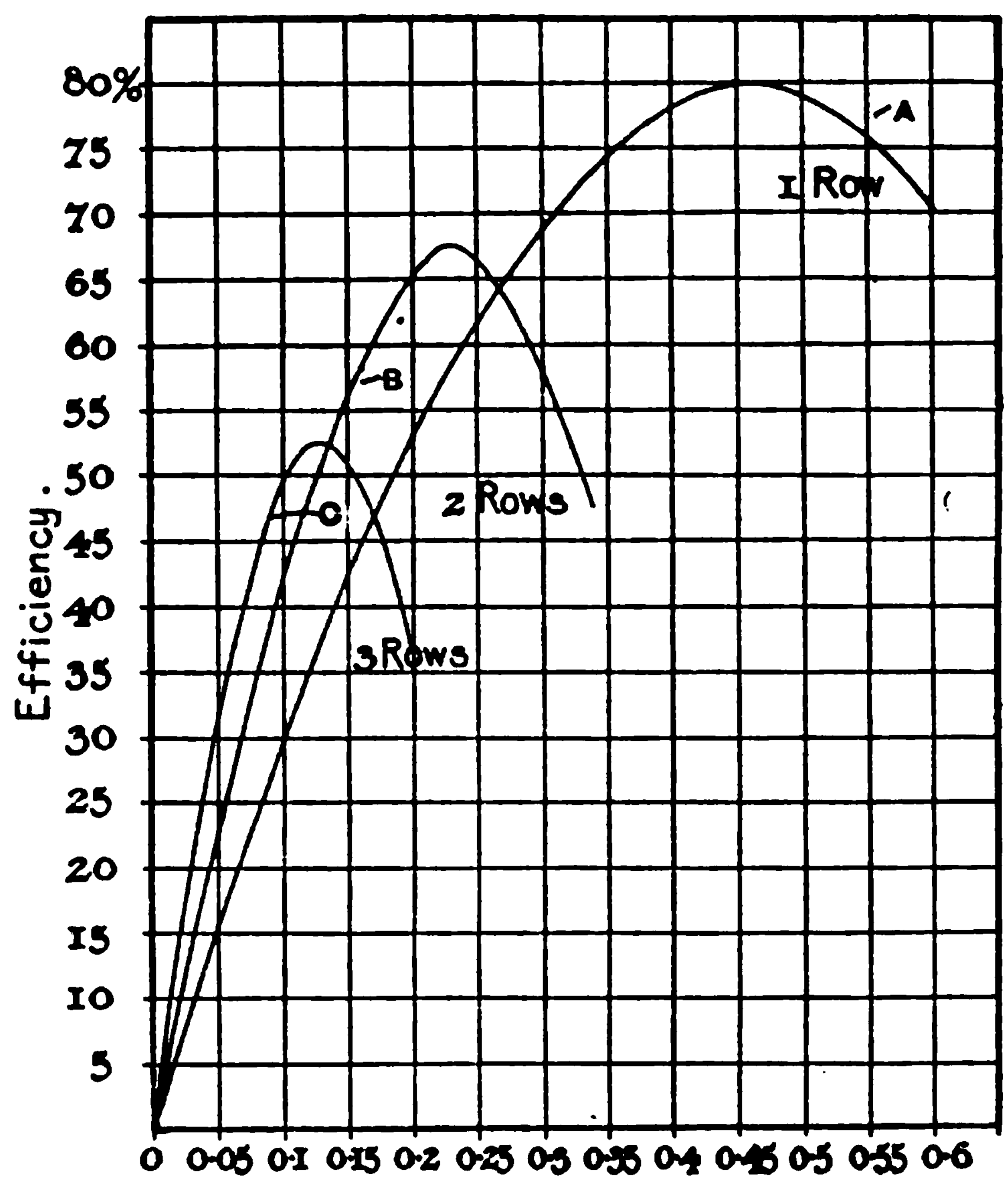
Type.	1. Reaction.	2. Impulse (Simple).	3. Impulse (Compounded for Pressure).	4. Impulse (Compounded for Velocity).	5. Impulse (Compounded for both Pressure and Velocity).	Combined Impulse and Reaction ("Drum and Disc.").	Radial Flow Reaction (Single Motion and Double Motion Types).
Construction	Drum fitted with moving blades, and casing fitted with guide blades.	Set of nozzles and impulse wheel with single blade row.	Consists of a number of nozzle stages, and corresponding wheels, each fitted with one blade row only.	A set of nozzles and one wheel fitted with from two to four blade rows, the casing being fitted with one row less of guide blades.	Sets of nozzle stages and wheels, each wheel being fitted with more than one blade row, and the casing also fitted with guide blades (one row less).	Usually consists of a velocity impulse wheel at the first stage with nozzles and moving blade rows, followed by a drum fitted with reaction blading.	Consists of round discs to which are secured concentric blade rings; steam enters at centre of inner blade ring, and exhausts at circumference of outer blade ring.
Example	Parsons.	De-Laval.	Curtis. Zoelly. De-Laval. Rateau.	Curtis. De-Laval. Rateau (British Westinghouse).	Curtis. De-Laval. Rateau (British Westinghouse).	Parsons. Brown-Curtis. British Thomson-Houston (Curtis). British Westinghouse (Rateau).	Ljungström.
Application	Marine propulsion.	Electric lighting sets, fans, etc. (geared-down).	Turbo-generators Electric light or power.	Electric light, etc. Marine propulsion.	Turbo-generators. Marine propulsion.	Turbo-generators. Marine propulsion.	Turbo-generators, etc., turbo-electric drives for marine propulsion.

For direct turbine drives in marine practice, Nos. 1 and 5 have been employed up to the advent of gearing and the use of superheat, since when Nos. 5 and 6 have been in use. The impulse turbine often fitted is that similar in design to No. 4, but having a number of pressure stages and only *one* velocity wheel at the first stage, the other stages consisting merely of nozzle discs and wheels with single blade rows. When double reduction gear comes into more general practice, turbines of the pure impulse type (turbo-generator class) will most likely be employed, as the newer conditions favour this design.

Principal Improvements.

Improvement.	Benefit Resulting.
(1) Single reduction gear, or double reduction gear	(1) High turbine speed, combined with low shaft speed. (2) Smaller and lighter turbines of impulse wheel type in place of drum reaction type. (3) Elimination of dummy rings.
(2) Use of Michell thrust block	(1) Total thrust taken up by a single collar, which is capable of carrying a pressure of 500 lbs. per square inch with safety.
(3) Superheat (100° to 150°)	Reduced water consumption per H.P. hour owing to the utilisation of the heat contained in the funnel gases.
(4) Improved design of blading, etc.	Reduced steam leakage loss over blade tops.

Efficiency of Blade Rows.



No. 4.—Efficiency of Blade Rows.

The diagram shows the relative efficiency of an impulse wheel fitted with 1 blade row, 2 blade rows, and 3 blade rows respectively.

6-Stage Impulse Turbine.

9. 6th stage diaphragm.
10. 1st stage impulse wheel (compounded for velocity).
11. 2nd " "
12. 3rd " "
13. 4th " "
14. 5th " "
15. 6th " "
16. Exhaust opening leading to L.P. initial end which is placed af

ow only of fixed blading which extends round the upper half casing

design to that of turbo-electric practice will be evident.

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The diagram, No. 4, reproduced from "Recent Developments in Steam Turbine Designs," by K. Baumann, shows the relative efficiency of one, two, and three blade rows respectively, when fitted on an impulse wheel. The ratio of blade speed to steam speed $\frac{Vt}{Vs}$ is also shown at the bottom of the diagram, and the results come out as follows for equal heat drop in each case:—

	One Wheel with Three Blade Rows (Curtis Wheel).	Three Wheels with Two Blade Rows each (Curtis Wheel).	Twelve Wheels with One Blade Row each (Rateau-De Laval, etc.).
Ratio $\frac{Vt}{Vs}$ - -	·133	·23	·46
Efficiency - -	52·5 per cent.	67·5 per cent.	77 per cent.

From the foregoing it will be evident that increasing the number of blade rows per wheel, and thus reducing the number of wheels, results in reduced efficiency. An example of this is found in the astern turbines, which usually consist of a single stage wheel fitted with three moving blade rows; also in the first stage of ahead turbines, which usually consist of a velocity wheel with two or three moving blade rows.

"Velocity" or Curtis Wheel.

The impulse wheel arranged with two or more moving blade rows, and usually fitted at the first stage of impulse turbines, is the invention of Curtis, and is commonly termed a "velocity" or "Curtis" wheel, for the reason that the velocity of the steam is extracted in steps in place of in one direct drop, and thus allows of lower shaft revolution speed, while yet retaining a suitable total ratio of steam speed to blade speed for reasonable efficiency results. The number of stages is also reduced, and the length of the turbine proportionately shortened.

Comparison of Velocity Wheel and Simple Impulse Wheel.

For similar efficiency results three impulse stages, each consisting of one nozzle diaphragm and wheel, are equivalent to one velocity wheel fitted with three moving blade rows and two fixed blade rows.

If, therefore, a considerable heat drop has to be allowed for in a single stage (as, for example, in astern turbines) a velocity wheel requires to be fitted to maintain the necessary value of $\frac{Vt}{Vs}$ throughout the stage. By reducing the velocity in a series of steps (say three) the sum of these will be equal to the total velocity drop required.

If, however, in the case of astern turbines which are only occasionally in use, maximum efficiency and economy are not of first importance, the ratio of V_{f} to V_{s} is usually of a much lower value than that of the ahead turbines of the same set, so that the full astern power may be thus obtained by means of a single stage velocity wheel turbine, which occupies only a minimum of fore and aft space in the vessel: this arrangement represents the latest practice in astern power.

The following extensive excerpt from the address of Mr Cleghorn exhaustively summarises the position of the modern type geared down turbine (also that of the near future) in comparison with other types, and the information contained is to be specially commended to the attention of students of marine turbine engineering, Tables No. I., II., and III., being of particular interest and value.

"But gearing in its rapid advance has hardly halted at the single spur and pinion drive. We now have the double arrangement coming largely into use, as the outcome of the success of the original form. This apparently minor development is obviously the special demand of the slow-speed ship, wherein the reduction ratio for suitable revolutions is very large; but it is not to be supposed that its utility is confined to the low-power low-revolution equipment. It can also with considerable gain in weight and efficiency be applied to high-powered vessels, as we shall see immediately. The great improvement due to double gear is mainly derived from its reaction on the permissible propeller revolutions and from the turbine equipment, the different elements of which can be arranged for their respective limiting speeds, and form compact units of insignificant dimensions compared with those of earlier days.

"In coincidence with all these changes, there has been a marked revival in the marine use of superheated steam. The continuous improvement in economy has led to a continual effort after further gain—the usual effect of successful thrift—and the diffidence in facing the troubles of high temperature steam has vanished before the bright prospect of augmented profit. In land power-plants, engineers have successfully used relatively high superheats, and are now tentatively approaching extreme high-pressure high-temperature conditions. The marine engineer is more seriously restricted, however, and a less riotous advance will require to satisfy him.

"And so we find, all well within this decade, the reciprocating engine and the direct turbine—separately, and in combination—achieve the zenith of their excellence, become stayed, and then supplanted by the rapid development of the geared transmission system; which, whilst induing the turbine with new powers and perfections, incidentally renders it a more appropriate vehicle for the safe carriage of that vexatious but profitable medium—superheated steam. The latest achievement of this period of restless progress in steam-propelling machinery is the superheat-geared-turbine equipment, transcending its forerunners in practically all respects, and

marking an epoch of advance without equal in the history of marine engineering.

"A consideration of this advance on an arithmetical basis, by an examination of tabulated average figures for different types of vessels, may therefore be instructive.

"These are given in Table I. For all types case (*a*) denotes standard designs of 9 or 10 years ago; (*b*) and (*c*) are present geared proposals with saturated steam and with 100° Fahr. superheat respectively; and while (*d*) is only so far actually utilised in type 4, it has been embodied in the others because of its possible and very probable application thereto.

"The figures are based on the reaction type of turbine, but roughly similar results may be expected for other systems. As will be seen, no consideration is taken of the hydraulic and electrical forms of transmission, for, apart from the difficulty of obtaining reliable figures for these, their practical test has been less extensive and severe, and consequently they are not yet on the same level of proved excellence.

"It must be understood that, since actual construction varies with service conditions, these figures only represent fair averages; but the guidance they provide may be considered sound.

"The relative fuel figures given are on the assumption of coal-fired boilers, and relate the fuel quantities under full power running. The use of oil offers many advantages, but to introduce figures for oil-fired boilers would necessitate the employment of fuel prices, and in a general table dealing only with types, they have been omitted. In fact, the adoption of coal or oil must be settled for each ship separately, and involves as main factors the fuel stations available on trading routes and the relative prices thereat, the gain in cargo space, freight rates, and boiler-room expenses. In general it may be said that oil prices rule too high to make the same overall economy with coal possible.

"The advantages derived from the various drives—ranging from the vanishing to the appearing forms—are similarly graded for all types, though to slightly varying degrees.

"The extreme design for the ocean liner shows a fuel gain of 25 per cent. over the 'obsolete.' Cases (*a*) and (*c*) in this scheme are from actual ships, and the difference therein exhibited may be accepted as clearly demonstrated. Case (*d*) may be questioned as being too extreme in this application; but in reality a double-geared superheated-steam arrangement was one of the original proposals for a large liner now on order, but so far as I am aware there is no case yet of definite procedure.

"At first sight a twin-screw double-gear set may not seem quite practicable for high powers, on account of the size of the turbines necessary to pass the steam. But it allows of the lowest propeller revolutions while the different turbine units may be arranged for their essentially different highest speeds. It is mainly the low-pressure turbine revolutions that are limited by the power, but this element

Table I.—Comparative Figures for Different Steam Machinery Types.

Type of Ship and Machinery. S = screw. S.S. = single screw.	Relative Fuel Consumption.	Relative Machinery Weight.	Relative Superficial Areas.	
			Engine Room.	Boiler Room.
1. Fast Ocean Liner.				
(a) 4 S. triple series direct-coupled turbines; cylindrical boilers—saturated steam	100	100	100	100
(b) 2 S. single geared turbines; cylindrical boilers—saturated steam	89	93	90	92
(c) 2 S. single geared turbines; cylindrical boilers—100° F. superheat	84·5	93·5	90	86
(d) 2 S. double geared turbines; cylindrical boilers — 200° F. superheat	75	88·5	86	77
2. Intermediate Liner.				
(a) 3 S. combination machinery; cylindrical boilers — saturated steam	100	100	100	100
(b) 2 S. single geared turbines; cylindrical boilers—saturated steam	93	80	87	92
(c) 2 S. single geared turbines; cylindrical boilers—100° F. superheat	88	81	87	91
(d) 2 S. double geared turbines; cylindrical boilers—200° F. superheat	74	72	83	80·5
3. Cross-Channel Steamer.				
(a) 3 S. compound direct-coupled turbines; cylindrical boilers—saturated steam	100	100	100	100
(b) 2 S. single geared turbines; W.T. boilers—saturated steam	91·5	75	100	97
(c) 2 S. single geared turbines; W.T. boilers—100° F. superheat	86·5	76	100	97
(d) 2 S. double geared turbines; W.T. boilers—200° F. superheat	73·5	68	95	86
4. Cargo Steamer.				
(a) S.S. triple-expansion reciprocating engine; cylindrical boilers — saturated steam	100	100	100	100
(b) S.S. single geared turbines; cylindrical boilers — saturated steam	83·5	79	85	95·5
(c) S.S. single geared turbines; cylindrical boilers—100° F. superheat	79	79·5	85	94
(d) S.S. double geared turbines: cylindrical boilers — 200° F. superheat	66	72·5	80	87·5

could be made double-flow or adapted for pinion contact direct on the secondary gear wheel. Extremely high powers could be attained, of course, by adding to the number of shafts.

"With the change from direct turbines to geared, there is a decrease both in boiler room and in engine room weights and spaces. With 100° Fahr. superheat there is no further gain, the superheaters and slightly heavier machinery counteracting the effect of reduced boiler capacity. But with higher degrees of superheat the boilers have a greater influence on the overall weights; and if double gears are adopted there is a small gain in the engine room accompanying the large effect of the improved consumption on the boilers.

"The 'obsolete' type for the intermediate class of liner is the well-known combination system. This has a distinctly better efficiency than the direct turbine, and the introduction of the geared system does not show the percentage improvement of the last case. But here we find the double gear becoming more distinctly suitable, owing to the increased range of turbine and propeller revolutions. The fuel gain of 26 per cent. from (*a*) to (*d*) for class 2 represents more than the corresponding 25 per cent. for class 1, the basis figure of 100 representing a higher order of excellence in the former than in the latter. The rapid downward trend of the machinery weight is here very noticeable, owing to the heavy type taken as the criterion.

"In channel vessels the driving system recently employed has been the three-shaft compound arrangement of direct-coupled turbines, with the high-pressure portion on the centre shaft, and parallel low-pressure units on the wing shafts. Space limitations have always dominated this turbine design, with the result that these were never very efficient examples of their type. The geared systems, again, show notable advantages, and are now always proposed. In this connection it is interesting to recall the historical case of the 'Normannia' and 'Hantonia,' pioneers of the geared drive for ships of this class. The success of that application, together with that of the famous 'Vespasian,' provided the initial impulse for the rapid development which has been the noteworthy feature of the past five years. The double reduction system has not the same absolute virtue as in the previous cases, since much lower propeller revolutions than in the case of the single gear scheme are not available; the lower limit for these being imposed by propeller dimensions. Nevertheless, the relative gain is immense because of the poor standard established by the original arrangement.

"It is now customary to fit these ships with water-tube boilers on account of weight considerations, and while these are somewhat less efficient than the cylindrical type, the saving in machinery weight is of superior value. This saving, however, does not seem to be accompanied by much diminution in the space occupied, and this point may frequently affect the superheat proposal.

"We come, lastly, to the very important type in merchant shipping, the low-speed ship of class 4. The displacement of the original

reciprocating engine, with all its excellent qualities, proved too much for the turbine alone, and it had to await the perfecting of the transmission system. The supremacy of the reciprocator has now been successfully challenged, and we must anticipate a gradual extinction of the old type as the most suitable machinery for modern cargo vessels.

"In the case of the single-geared arrangement without superheat, an improvement in fuel consumption of 16½ per cent. is attainable. This order of superiority is quite certain, having been clearly demonstrated in a variety of cases. The augmented advantage obtained by even the lower degree of superheat is also very definite, and with this class of ship superheat makes a quite unambiguous claim.

"But this single-geared arrangement is on the point of being divested of its brief supremacy by the further development of its own species. The double gear is for this case pre-eminently applicable, since the limiting revolutions of the turbine, at the low powers required, are of an exceedingly high order compared with the best revolutions for the propeller. Thus, supposing we take the latter at 70 per minute, a single gear with large reduction ratio would still only allow of 1300 to 1400 revolutions per minute for the turbine, and both high-pressure and low-pressure units would require to run at the same speed. Higher turbine revolutions could only be obtained by increasing the propeller speed, and the advantage of such an increase is not clear. But, by using a double-gear, with propeller revolutions at 70 per minute, we could have a three-pinion arrangement of primary drive with speeds of the order of roughly 4000, 3000, and 2000 revolutions per minute respectively, the two former transmitting ¼ of the power each, and the latter ½ of the power.

"The benefits are manifold. The propeller efficiency is maintained at a high value, and the generation of a minimum power takes place in extremely efficient turbine units, small in dimensions and compact in construction, and thus of great suitability for superheated steam. These advantages are necessarily accompanied by positive gains in fuel, weight, and space, large even over those of the prototype, and immense compared with the standard set up by that old system that has done, and is yet doing, such splendid service.

"Although of recent application, the great appeal made by the superheat-double-gear installation is clearly reflected in the increasing number of cargo ships, building or on order, to be equipped with this system. The advantages are yet anticipatory, but the ardent endeavour to attain realisation thereof is indicative of a profound belief in their promise; and the actual performance of the new ships, when they have been well tried on service, will be studied with great interest.

"Throughout Table I. it is assumed that the steam auxiliaries in use are working with the superheat arranged for the turbines. It is occasionally the practice to run the main engines only on superheat, and to provide facilities for supplying the auxiliaries with saturated

steam. This cannot be accepted as a final solution, nor is it quite general, and, for purposes of comparison within a stage of progress, we are justified in assuming that, where superheat is adopted, it will be used throughout the engine room.

"With regard to the ratios of water consumptions underlying the estimates, of which Table I. gives a synopsis in relative figures, it may be desirable to add, that where experience has been gained with the actual ship—*e.g.*, case (*a*) for all classes—the water consumptions taken are those on a long full-power trial.

"Cases (*a*) and (*c*) for the ocean liner class are from actual ships, and I subjoin these figures.

TABLE II.—WATER CONSUMPTION FOR LARGE LINERS.

Machinery Type.	S.H.P.	Steam Condition at Turbines.	Lbs. per S.H.P. per hour.		
			Turbines only.	Other Purposes.	All Purposes.
4 S. direct-coupled triple series turbines with parallel L.P.'s on inner shafts	23000	Saturated	11·2	2·2	13·4
2 S. single geared compound turbines with 1 H.P. turbine and 1 L.P. turbine per set	14500	85° F. Superheat	9·7	2·3	12·0

"The auxiliary consumption may be somewhat different on service, but the difference will affect the various cases similarly, and could have no serious influence on overall comparative figures.

"With double gearing there is a further gain in turbine consumption, the main reasons for which I will refer to later, the relative figures for the geared arrangements being approximately as follows:—

TABLE III.—RELATIVE TURBINE CONSUMPTION FOR GEARED ARRANGEMENTS.

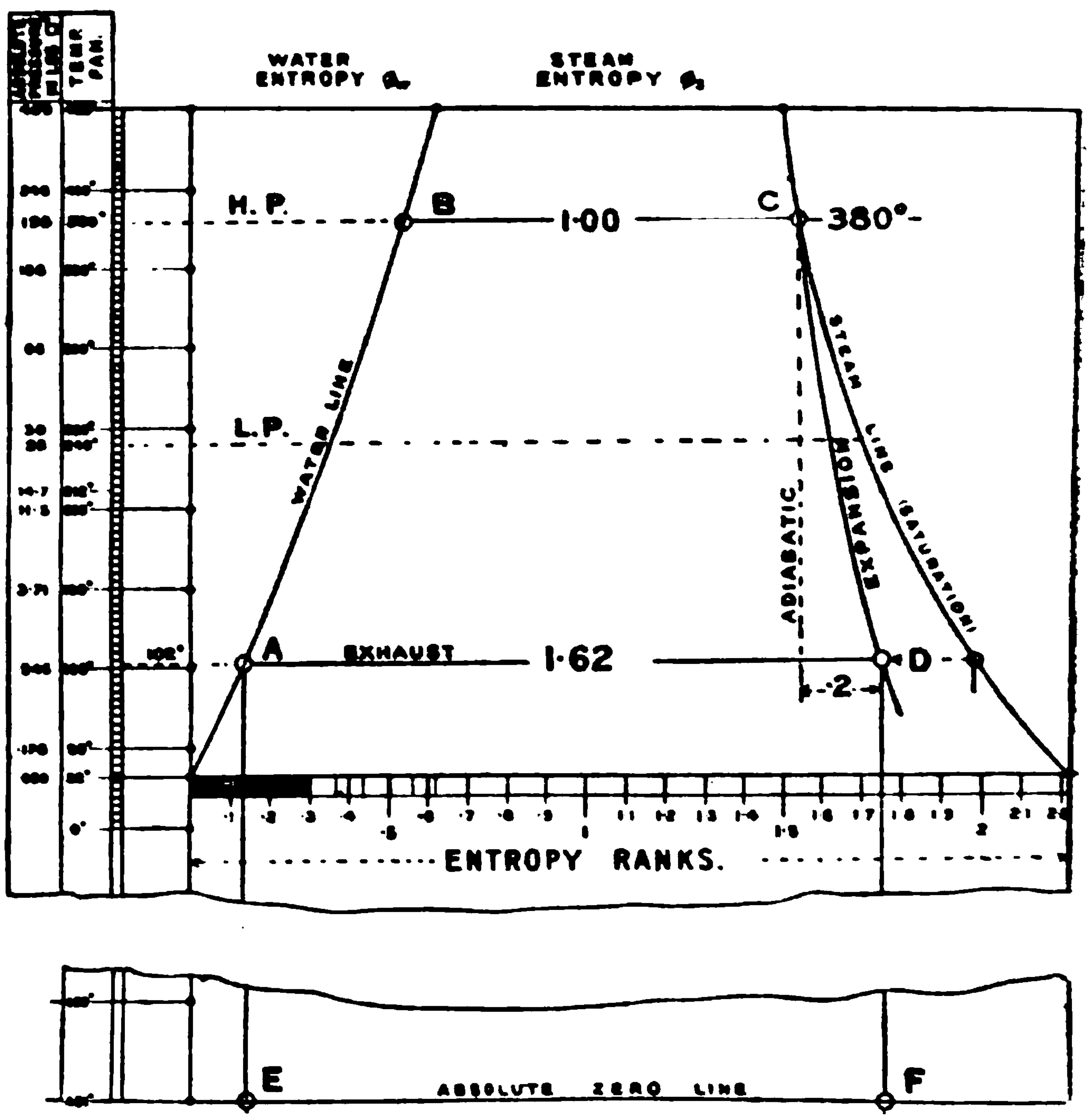
Geared Type.	Steam Condition.	Relative Consumption per S.H.P. per hour. Turbines only.
Single reduction gearing	Saturated	100
Double reduction gearing	,,	91
,, ,, ,,	100° F. superheat	82·5
,, ,, ,,	200° F. ,,	74·5

"These figures apply very directly to the low-speed type in class 4 of Table I., and while they may vary somewhat with other

classes, the gain in propeller efficiency made possible by the adoption of double gearing in the higher speed vessels must be emphasised."

Improvement in Efficiency Due to Superheat (by Entropy Diagram).

EXAMPLE.—Compare the heat efficiency ratio of saturated steam at 180 lbs. gauge, and of saturated steam at 180 lbs. gauge, with a superheat of 150° Fahr., the exhaust pressure in both cases being



No. 7. Entropy Diagram for Saturated Steam of 180 lbs. Gauge and Exhaust 1 lb. Absolute.

1 lb. absolute, and 102° Fahr. Assume ·2 entropy to right of adiabatic line as the turbine expansion line, due to the reheating effect of blade tip leakage.

Saturated Steam.

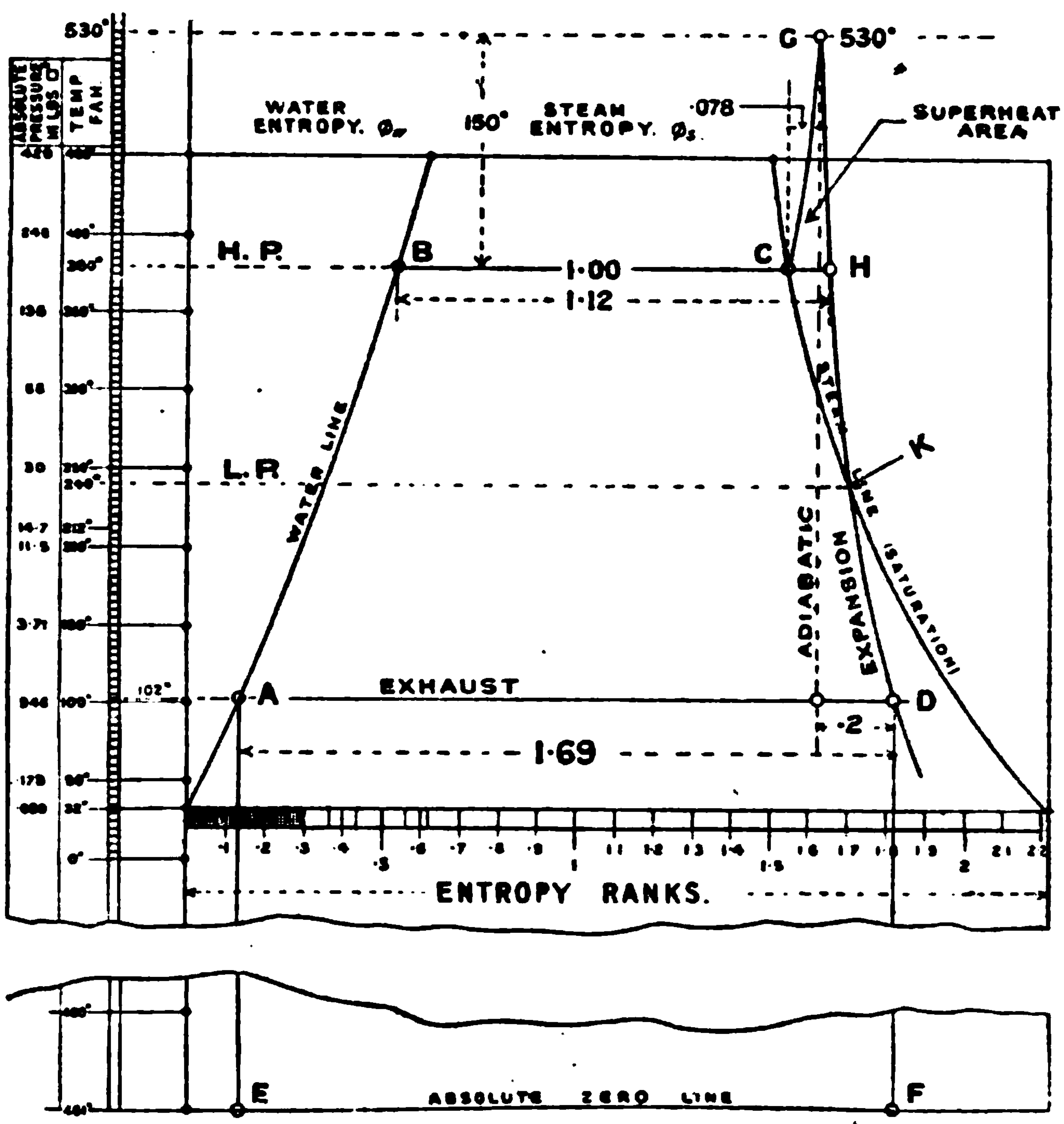
First draw horizontal lines at the positions of the temperature levels corresponding with the absolute pressures, 195 lbs. and 1 lb. (B, C, and A, D). Then, 180+15=195 lbs. absolute, the temperature of which is 379·7° Fahr. (say 380°) from steam table. Again, 1 lb. absolute=102° Fahr. from steam table. To obtain point D measure off ·2 entropy to right of adiabatic line, and draw in the curve of

turbine expansion from C to D. Measure the entropy on each line, which will be 1·00 at the 380° Fahr. level (B, C), and 1·62 at the 102° Fahr. level (A, D).

Work done area = A, B, C, D.

Heat supplied area = E, A, B, C, D, F.

Then, **Heat efficiency ratio** $= \dfrac{A, B, C, D}{E, A, B, C, D, F}$



No. 8. Entropy Diagram for Superheated Steam at 180 lbs. Gauge and Exhaust 1 lb. Absolute.

$$\textbf{Work done in B.T.U.'s} = \left(\frac{1{\cdot}00 + 1{\cdot}62}{2}\right) \times (380° - 102°) = 364{\cdot}18 \text{ B.T.U.}$$

$$\textbf{Heat supplied} = (102 + 461) \times 1{\cdot}62 + 364{\cdot}18 = 1276{\cdot}24.$$

$$\textbf{Efficiency ratio} = \frac{364{\cdot}18}{1276{\cdot}24} = {\cdot}285.$$

Superheated Steam.

$$\textbf{Rule, Entropy} = {\cdot}48 \times \left(\frac{\text{Superheat}}{\text{Mean absolute temperature}}\right).$$

First draw a line at the temperature level of the steam when superheated, which = 380° + 150° = 530° Fahr.; next, calculate the entropy as shown by the rule :—

Then, $380° + 461° = 841°$ absolute,

And, $380° + 150° + 461° = 991°$ absolute.

$$\text{Mean absolute temperature} = \frac{841° + 991°}{2} = 916°.$$

$$\text{Entropy} = \cdot 48 \times \left(\frac{150°}{916}\right) = \cdot 078.$$

Measure off ·078 entropy to right of C, and run up the dotted line until it strikes the 530° temperature level at point G, and which, if drawn downwards, forms the adiabatic expansion line. At temperature level of 102° Fahr. measure ·2 entropy to *right* of the new adiabatic line so found (as in the previous case), and draw in the approximate turbine expansion line from G to D. The curve C, D gives a position H at the 380° Fahr. level, and the increase of entropy at this point, if measured, will be found to be ·12, which now gives the entropy value as 1·12 in place of 1·00 at 380° temperature level.

$$\text{Work done area} = \text{C, G, H} + \text{A, B, H, D.}$$

$$\text{Heat supplied area} = \text{C, G, H} + \text{A, B, H, D} + \text{E, A, D, F.}$$

Then, $$\text{Heat efficiency ratio} = \frac{\text{C, G, H} + \text{A, B, H, D.}}{\text{C, G, H} + \text{A, B, H, D} + \text{E, A, D, F.}}$$

$$\text{B.T.U. in area C, G, H} = \left(\frac{0 + \cdot 12}{2}\right) \times (530 - 380) = 9.$$

$$\text{B.T.U. in area, A, B, H, D} = \left(\frac{1 \cdot 12 + 1 \cdot 69}{2}\right) \times (380 - 102) = 390.5.$$

$$\text{Total B.T.U. (work done)} = 390 \cdot 5 + 9 = 399 \cdot 5.$$

$$\text{B.T.U. supplied in area E, A. D, F} = 1 \cdot 69 \times (102 + 461) = 951 \cdot 4.$$

$$\text{Total B.T.U. supplied} = 399 \cdot 5 + 951 \cdot 4 = 1350 \cdot 9.$$

Then, $$\text{Efficiency} = \frac{399 \cdot 5}{1350 \cdot 9} = \cdot 296 \text{ (nearly).}$$

Then, Efficiency of saturated steam $= \cdot 285$.

And, Efficiency of superheated steam $= \cdot 296$.

$$\text{Per cent. increase} = \frac{(\cdot 296 - \cdot 285) \times 100}{\cdot 285} = 3 \cdot 8 \text{ per cent.}$$

In the above example it will be noted that the affects of the superheat fall off to zero at K, the end of the last H.P. turbine expansion.

Superheat.

The practical benefits of superheating, as proved by tests, exceed the theoretical gains as calculated by the entropy diagram. For a superheat of 150° the gain in efficiency works out as 3·8 per cent., as shown by the example; whereas, according to K. Baumann, the actual gain in efficiency amounts to fully 5½ per cent.

Superheat Practice.

The use of superheated steam is now common in all turbine practice, the amount of superheat varying in degree and ranging from 100° to as high as 300° Fahr. in the case of Ljungström turbines.

The economy of superheat, according to K. Baumann, is as follows:—

Between—

0°-100° F. superheat, 1 per cent. improvement in steam consumption for every 10° F. superheat.

100°-200° F. superheat, 1 per cent. improvement in steam consumption for every 12° F. superheat.

200°-300° F. superheat, 1 per cent. improvement in steam consumption for every 14° F. superheat.

The improvements in turbine efficiency resulting from superheat are given as follows:—

4·25 per cent. better efficiency at 100° F. superheat than for dry saturated steam.

675 per cent. better efficiency at 200° F. superheat than with dry saturated steam.

7·5 per cent. better efficiency at 300° F. superheat than for dry saturated steam.

Adoption of Impulse Turbines.

As mechanical gearing down has made higher turbine speeds permissible, turbines of various other designs will now be able to compete with the Parsons and Curtis, which, until quite recently, held the market for marine practice. It may shortly be expected, therefore, that turbines of the Rateau, Curtis, and other types will be adopted and installed for trial in vessels, and thus enter into keen competition with the original Parsons' reaction type, which has for so many years occupied the premier position in the field of marine practice.

Comparison of Low and High Speed Turbines.

Curtis turbines, as formerly constructed for direct propeller drives, consisted of a number of pressure stages (from five to seven) and velocity stages, the latter formed of from either three or four moving blade rows fixed to the impulse wheel, and two or three fixed blade rows secured to the casing, this construction being necessary to reduce the velocity of the steam down to suit the shaft speed, and to bear a certain ratio to the same for good efficiency.

Curtis turbines as at present constructed for geared down drives usually consist of one velocity wheel at the first stage only, with two or three moving blade rows and one or two fixed blade rows, the following stages consisting merely of simple impulse wheels with one moving blade row, and therefore no fixed blade rows in the casing, diaphragm discs with suitable nozzle openings being sufficient for the purpose.

With this arrangement the speed of the steam is kept at a higher value, and in fact this design is similar to that followed out in

electrical practice where the "turbo-generators" run at high revolution speed per minute, and thus allow of higher steam speeds.

In the case of L.P. turbines the reaction drum stage is, in some cases, still fitted, but compared with previous practice, the total number of expansions is increased, whereas the number of blade rows in each expansion (velocity stages) is reduced: this holds good for both Curtis and Parsons' types.

Variation in Blade Sections.

The initial impulse stages are fitted with the usual section of blading shown in No. 43, page 397, but as the stages progress the blade sections change from what might be termed the standard impulse section to that of the Parsons section (No. 44, page 398). This arrangement allows better for the increased volume of the steam at lower pressure, and together with the increased pitch of blades (less number per row) and thinner section keeps down the blade heights to reasonable proportions. As also referred to elsewhere, the last three stages of the L.P. turbine are often fitted with blades of equal height, but of increasing angle, and of thinner Parsons-like section, in place of Curtis impulse section. Copper is now being more extensively employed in the composition of the material used for blading.

Blade Packers.—The packers (of brass in nearly all stages) are slightly different in section from the original Curtis type, being thicker on the exhaust side than on the admission side, the blades being shaped to correspond, which means that the latter lie at a slight angle axially, in place of being placed at dead right angles to the casing or rotor diameter. It might be said that the modern impulse stage blading is a combine of Parsons and Curtis blade sections, possessing features common to both, the resemblance to the Parsons section increasing progressively with the turbine stages from the initial to the exhaust end.

In certain cases (short H.P. blades or low-pressure stage L.P. blades) separate blade packers are omitted, the spacing being allowed for by thickening the blade at the root sufficiently to act as a solid packer. In all cases the caulking is of the circumferential method, the grooves for blades and packers being undercut as described previously (p. 447).

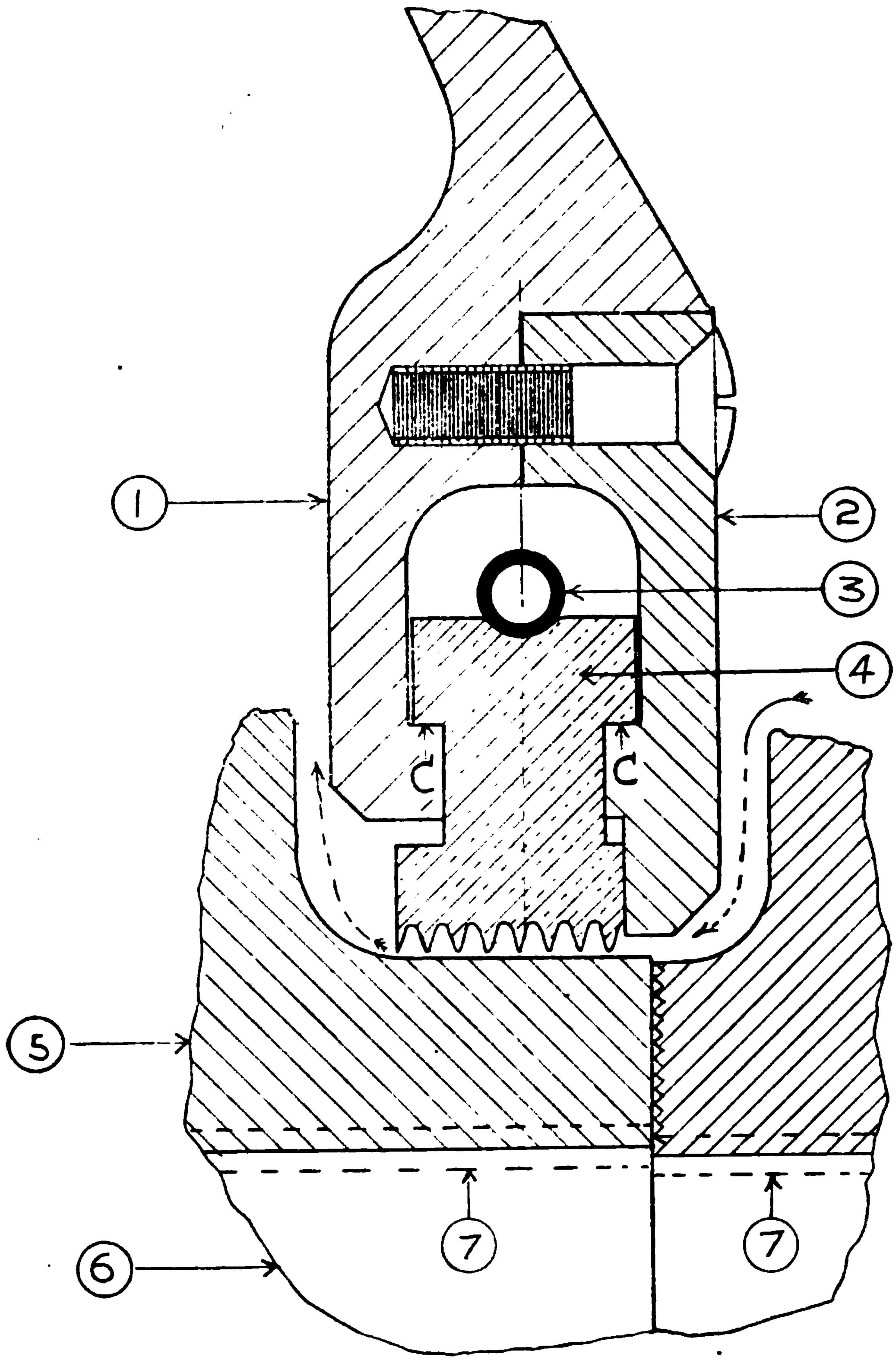
Nozzle Diaphragm Glands.

While it is usual to open up both sides of a nozzle stage by ports cut in the wheels to equalise the pressure on either side and reduce blade tip leakage, it is also as important to prevent leakage between pressure stages past the nozzle round the shaft diaphragm discs where they fit. This, however, has always been found a matter of some difficulty from the beginning, and many experiments and trials have been carried out in recent years with a view of overcoming this mechanical trouble, peculiar in itself to impulse stage turbines. Up to the present, perhaps the most effective form of packing is that shown in the

No. 9.—Nozzle Stage Diaphragm.

1. Nozzle openings.
2. Brass ring in three segments.
3. Cover plate.
4. Cut in ring.

The segmental brass ring is pressed lightly against the hub of the impulse wheel by means of a light garter spring, and the serrations or teeth act to wire-draw any leakage steam which may tend to pass from a higher to a lower pressure stage. The ring is practically a "touch" fit, as only about ·005" clearance is allowed when cold.



No. 10.—Impulse Stage Diaphragm Packing.

1. Diaphragm boss.
2. Cover plate.
3. Garter spring.
4. Bronze ring in three segments, and bearing at C.C. when at limit position.
5. Hub of wheel.
6. Taper on shaft (1 in 24).
7. Key.

The arrows show the direction of leaking steam flow.

The serrations shown are usually arranged to set up a "touch fit" when running, the light spring compression allowing of an elastic action and eliminating risk of damage by actual metallic contact.

sketch (Nos. 9, 10), and which is fitted in Curtis type turbines. The bronze ring shown is fitted in three segments, and is serrated or toothed, thus forming a kind of labyrinth packing, the leaking steam being alternately wire drawn and expanded from pitch to pitch on the teethlike projections (No. 10). When running, the projections of the serrations almost touch the revolving wheel bosses, a very fine clearance being allowed when cold about $\frac{20}{1000}$ in., or ·02 in.

High and Low Speed Impulse Turbine.

In direct drive Curtis turbines each impulse stage wheel is of the velocity type, leaving from three to four moving blade rows, with corresponding fixed blade rows in the casing, and this design is found necessary to reduce the steam velocity to meet the requirements of the slow speed of the turbine; but with gearing down this is not required, as with higher turbine revolutions, and, therefore, blade speed, the energy can be extracted by simple impulse stage wheels in which the steam velocity is also higher, hence the change in turbine design, of which the sketches (No. 6, facing page 701, and No. 11, facing page 714) are typical examples. It should be noted that for equal stage pressure drops a velocity wheel may be necessary for slow turbine speeds, whereas a single blade impulse wheel may be found sufficient for higher turbine speeds, and this condition maintains in the modern impulse turbine with geared down drives. The impulse wheel turbine is therefore of shorter length than the velocity wheel type, and as the revolutions are higher the diameter is also reduced, so that, for a given power, the turbines occupy much less machinery space than formerly.

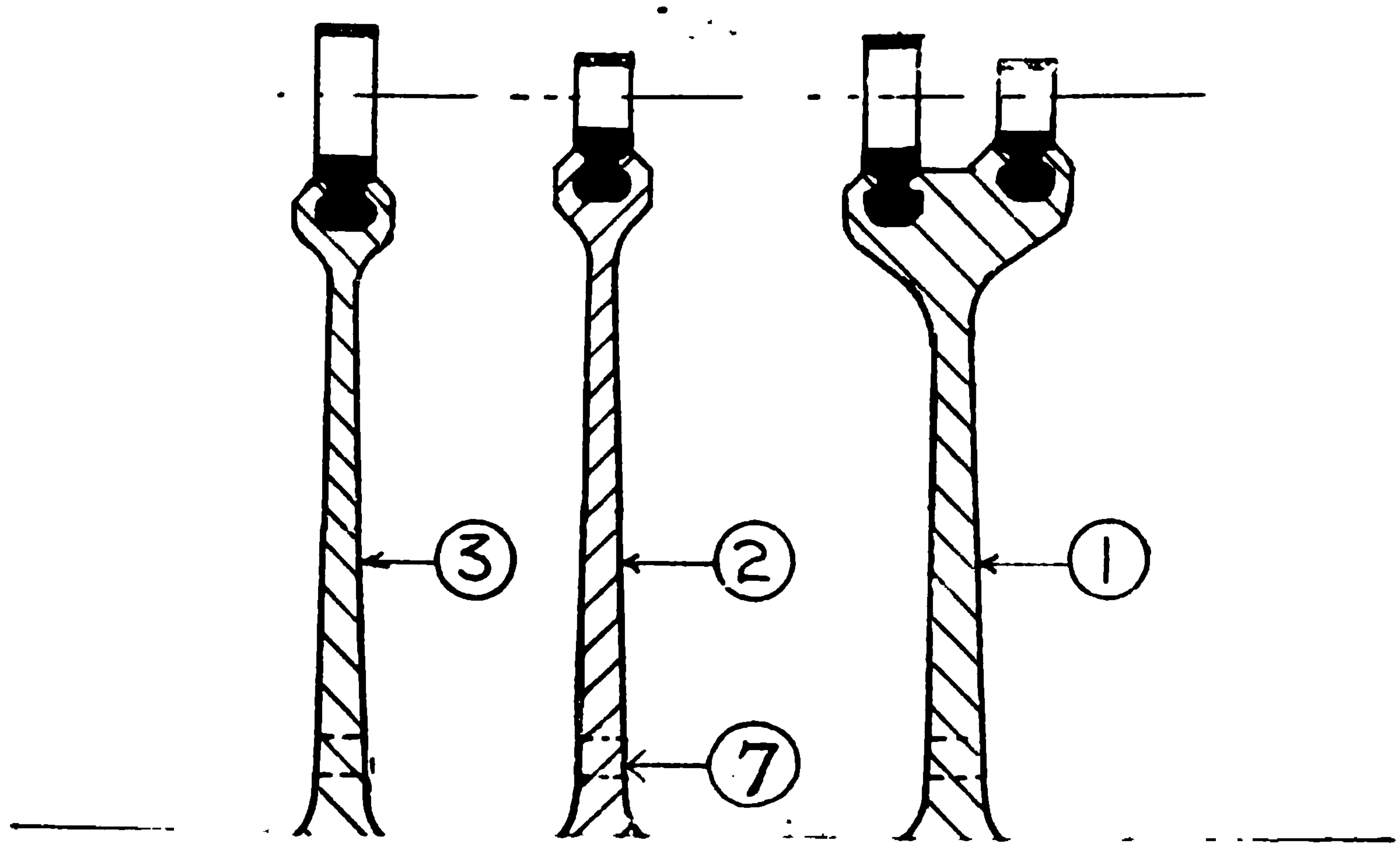
Blade Length and Mean Diameter of Pitch Circle.

In turbo-generator practice the mean blade diameter is usually kept constant throughout the turbine, which means that the wheels are less in diameter at each stage successively, but the diaphragms increase in diameter at each stage, the turbine casing therefore also increases in diameter towards the exhaust end (No. 46, facing page 548).

In marine practice the casings are often kept parallel, which means that the pitch circle diameter of the blades (p.c.d.) becomes less at each stage from the initial end to the exhaust end. Again, as is usual with reaction turbines, the last three expansions of the L.P. turbines have blade rows and nozzle openings of equal height, but the blade angle and blade pitch is increased to allow for the rapid increase of steam volume at the low vacuum pressures of these stages. In other words, each row contains fewer blades than the previous one.

The foregoing assumes that in the preceding stages the admission has been "full," that is all round the diaphragm circumference, which is usually the case in L.P. turbines.

If not, then the segmental area of nozzle openings may be increased to allow of the flow of the lower pressure steam, at succeeding stages, from initial to exhaust, and thus give additional annulus area.



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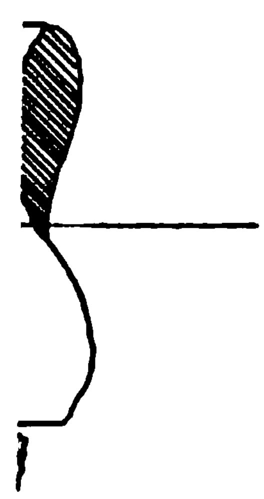

keep the
oil within the limits of the bearing, and also acts to prevent the ...ng gland steam from obtaining access to the bearing, which would result in heating up of the latter.

As the turbine glands and main bearings are in close proximity, it is essential that steam leakage from the gland carbons should be kept away from the bearings, otherwise a considerable rise of temperature of the bearings may gradually develop, due to the heat of the leaking steam. For this reason oil baffles are usually fitted

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SECTION XIX.

OTES AND SKETCHES RELATING TO GENERAL MARINE STEAM TURBINE PRACTICE.

l and Steam Baffles.

No. 1.—Main Bearing of Turbine.

1, 2. Oil baffles.
3. Rotating baffle.
4. Carbon gland rings.

The rotating baffle travels round with the shaft, and tends to keep the oil within the limits of the bearing, and also acts to prevent the leaking gland steam from obtaining access to the bearing, which would result in heating up of the latter.

As the turbine glands and main bearings are in close proximity, it is essential that steam leakage from the gland carbons should be kept away from the bearings, otherwise a considerable rise of temperature of the bearings may gradually develop, due to the heat of the leaking steam. For this reason oil baffles are usually fitted

on the bearings to keep the oil in its place, also a steam baffle of the rotating type is fitted on the shaft as shown in the sketch, to prevent the gland steam from reaching the main bearings.

Oil temperature in bearings should not be allowed to rise above 160° Fah.

Braking by Steam.

If when, say, running ahead the steam is shut off from the main turbines the shafts will still continue to revolve, owing to the "way" on the vessel; it may therefore become necessary to open up steam to the astern turbines to act as a brake, and so bring the vessel to a standstill.

The same will hold good when running astern, braking steam being then admitted to the ahead turbines to check the speed of the vessel.

Blade Pitch Circle Diameter.

If the turbine casing is kept parallel throughout its length, then the mean blade pitch diameter of each wheel blade then becomes less from initial to exhaust end, thus reducing the blade speed at each stage, but, as in impulse turbines, the steam speed is also reduced at each stage by the energy extracted, the ratio of V_r to V_s still remains fairly constant through the turbine.

Holes in Stage Wheels (No. 2).

The tapped holes shown in the various stage wheels serve for three distinct purposes, as follows:—

1. For drawing off the wheels in overhauling or repair.

2. For balance of pressure on either side of stage to reduce blade tip leakage.

3. For reducing the steam thrust on the turbine rotor (aft or forward as the case may be).

Oil in Turbines.

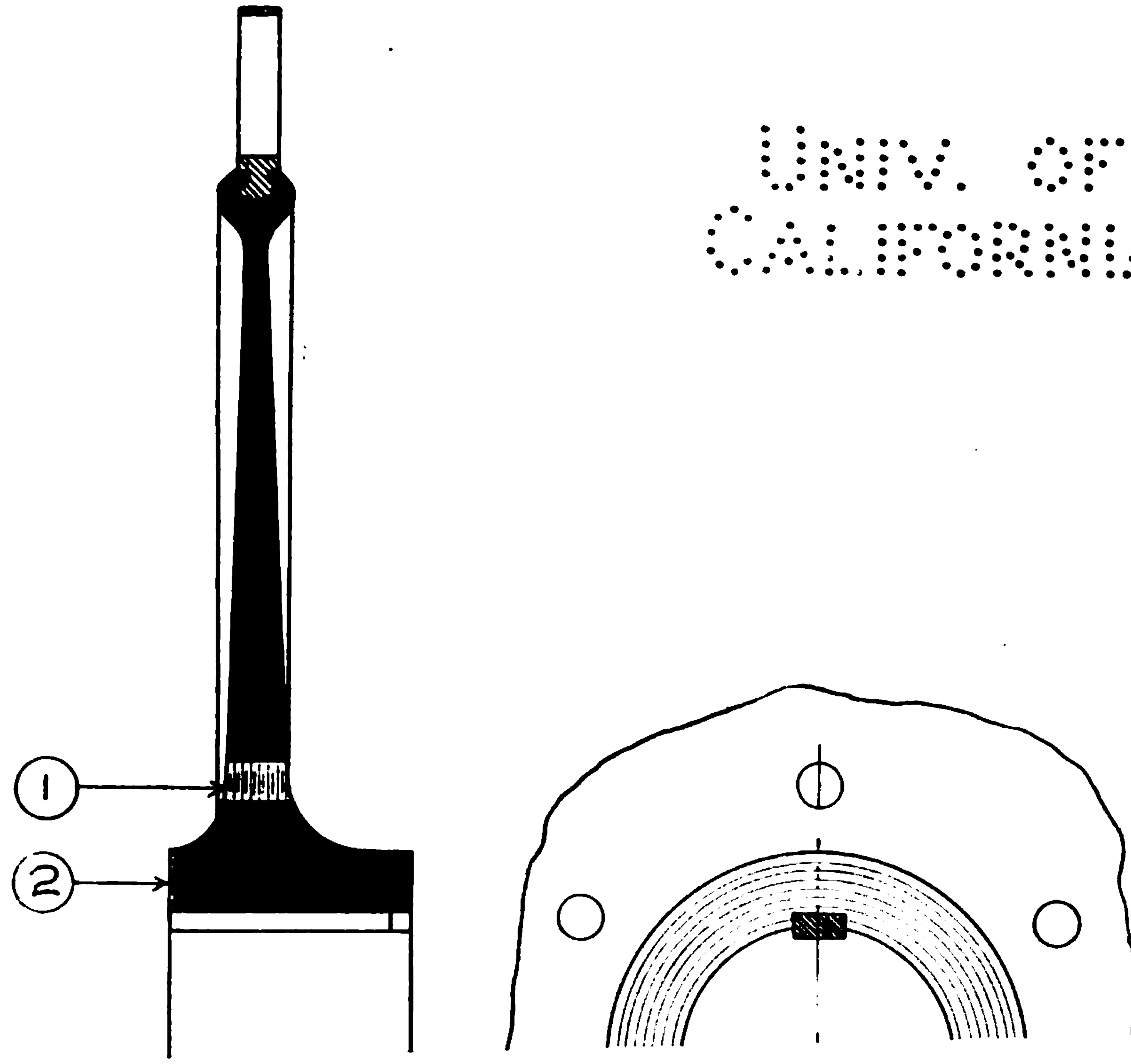
It need hardly be pointed out that if oil finds its way into the turbine various troubles may arise, such as the following:—

1. Danger of oil in boilers causing buckled plates, corrosion, priming, which latter would again likely produce blade stripping in the turbines. Also choking up of superheater tubes.

2. Damage to nozzles, turbine blades, etc., and possible corrosion in turbine rotor or casing.

Glands.

The carbon block system (No. 8) of gland packing, as originated in the Curtis design, is now coming more into general use for other designs, and appears to have stood the test of years with satisfactory results, although recent experiments have proved the existence of a "critical speed," above which rapid heating of the carbons takes place.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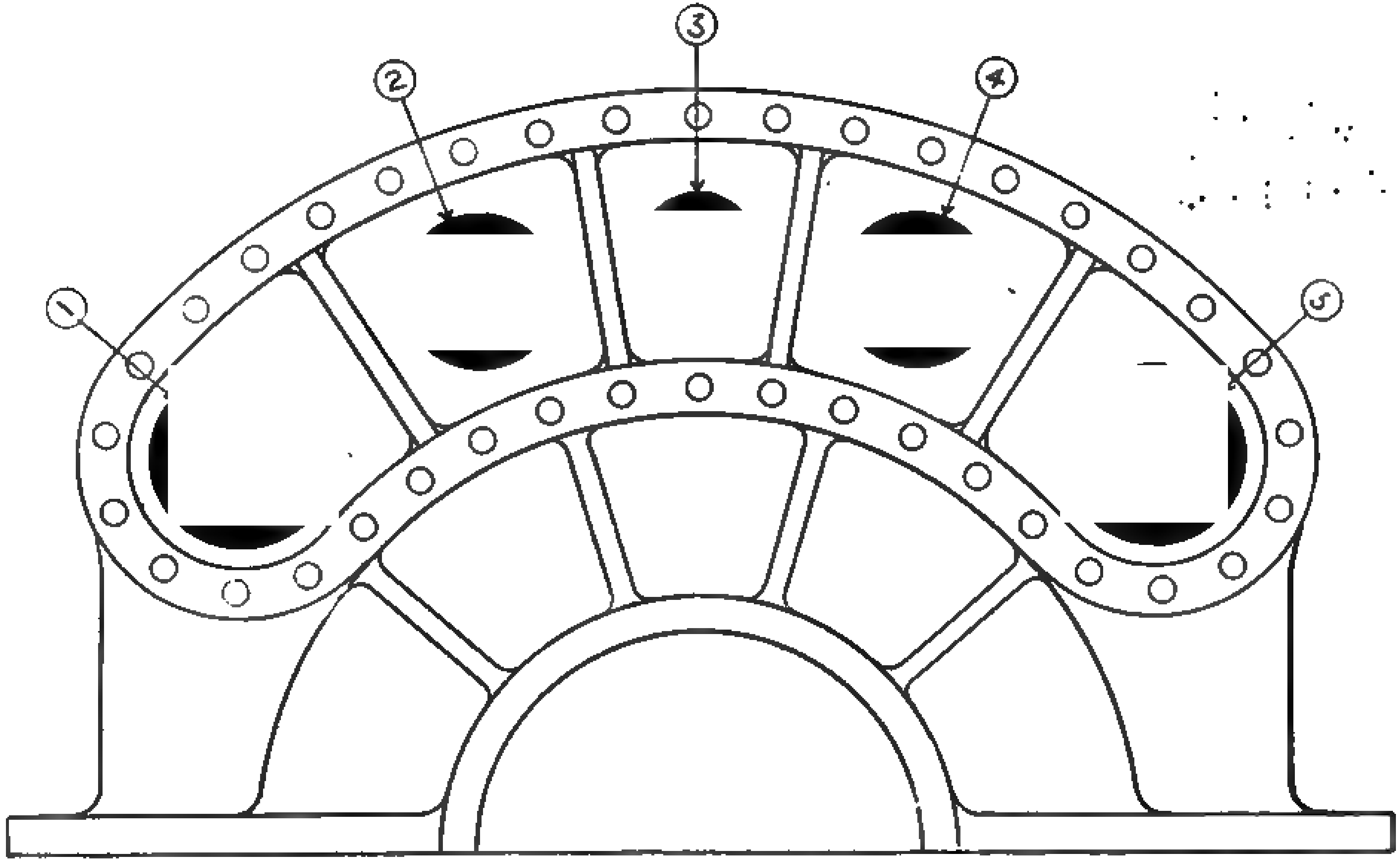
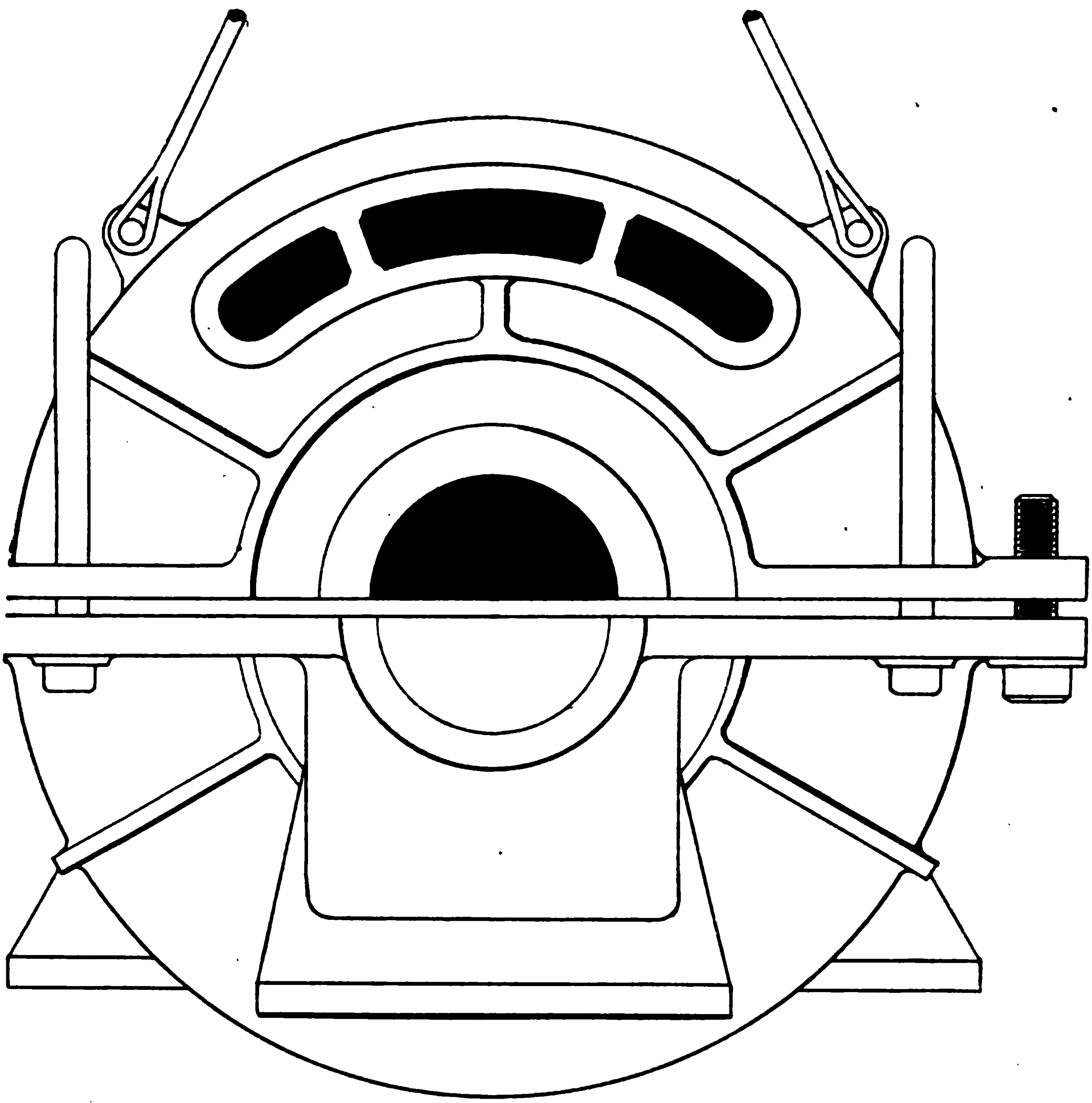


No. 5.—End View of Nozzle Control Box Openings of H.P. Turbine.

1. Opening for, say, 7 valves.
2. " " 4 "
3. " " 2 "
4. " " 5 "
5. " " 7 "

The openings shown are each controlled by a hand-regulating valve, and each opening gives steam to a certain number of nozzles, as shown above.



7.—View of Turbine—Upper Half Turbine Casing being Lowered into Position on to Bottom Half.

:heck the effects of "spring" due to weight, a few bolts are passed up through the holes before the flanges of the two halves come into contact.

Annulus Area.

By this is meant the area open for steam flow through the blade rings of a turbine, neglecting blade thicknesses.

The effective annulus area may be increased as follows :—

1. By lengthening the blades.
2. By thinning the blades and increasing the blade angle (fitting semi-wing or wing blades in the case of Parsons' turbine).
3. By reducing the number of blades per ring.

In practice the three systems mentioned are employed conjointly in both impulse and reaction turbines.

In the nozzles of an impulse turbine the annulus area open for steam flow can be increased at the earlier H.P. stages by extending the nozzle arc further round the periphery.

Annulus Factor.

The factor represents the effective area open for steam flow when the blade angle and thickness is allowed for, and it varies from ·33 for H.P. first stage blades, to about ·7 or more for full "wing" type blades as fitted on the last L P. expansions of Parsons' reaction turbines.

Blade Height Ratio and Blade Rows.

In high speed impulse type turbines the increase of blade height varies from about 1·2 in H.P. rotors to about 1·12 or 1·2 in L.P. rotors, while the number of blades per ring or row usually become less progressively from initial to exhaust end, although exceptions are occasionally made to this standard practice.

The following data is merely intended to give a general idea of the approximate run of blade heights and number of blades per row, etc., obtaining in a modern geared-down impulse turbine set, consisting of one H.P., one L.P., and one astern turbine. The H.P. and L.P. only are referred to in the roughly calculated data shown.

H.P. Turbine (Revs. 2500).

Stage.	Mean Dia. of Nozzle Openings and Blades.	Number of Blade Rows in Wheel.	Depth of Nozzles.	Blade Heights.	Number of Blades per Row.	Nozzle Opening Area.
1		2	$1\frac{1}{2}''$ and $1\frac{3}{4}''$	$2''$ and $3\frac{1}{8}''$	150 and 155	Partial Admission
2		1	$1\frac{3}{4}''$	$2\frac{1}{4}''$	230	Full Admission
3	30″	1	$2\frac{1}{4}''$	$2\frac{7}{8}''$	226	
4		1	$2\frac{7}{8}''$	$3\frac{3}{8}''$	220	”
5		1	$3\frac{1}{4}''$	$3\frac{7}{8}''$	205	”
6		1	$3\frac{5}{8}''$	$4\frac{1}{4}''$	200	”

To prevent "spreading" or "spilling" of the jet the blade heights are made in excess of that of the nozzles (about $\frac{1}{4}''$ or $\frac{3}{8}''$ overlap top and bottom).

H.P. and L.P. Annulus Area.

H.P. Annulus Area.

Diameter of sixth stage H.P. wheel $= 30'' - 4{\cdot}25'' = 25{\cdot}75''$.

Diameter over blades at sixth stage $= 30'' + 4{\cdot}25'' = 34{\cdot}25''$.

Then, **Annulus Area** $= (34{\cdot}25^2 - 25{\cdot}75^2) \times {\cdot}7854 = 461\boxed{''}$.

L.P. Annulus Area.

Diameter of first stage L.P. wheel $= 50''$.

Assuming equal volumes of steam for H.P. terminal pressure and L.P. initial pressure, and therefore equal annulus areas required,

Then, $(34{\cdot}5^2 - 28{\cdot}75^2) = (D^2 - 50^2) = 461 = (D^2 - 50^2)$.

Then, $D^2 = 461 + 50^2 = 2961$.

And, $\sqrt{D} = D = 54{\cdot}5''$ **diameter over blades.**

Blade Height $= \dfrac{54{\cdot}5'' - 50''}{2} = 2\frac{1}{4}''$ **for equal annulus area.**

If however, a blade height ratio of say, 1·2 be assumed to allow for pressure drop and increase of volume in the L.P. receiver,

Then, **Actual blade height at first L.P. stage** $= 2{\cdot}25'' \times 1{\cdot}2 = 2{\cdot}7''$, **say** $2\frac{3}{4}''$.

The remaining blade heights are worked out from the same blade height ratio figure of 1·2.

As in the case of the H.P. turbine, the blades overlap the nozzle openings at top and bottom to reduce "spreading" of the steam jet.

L.P. Turbine (Revs. 2500).

Stage.	Meam Dia. of Nozzle Openings and Blades.	Number of Blade Rows in Wheel.	Depth of Nozzles.	Blade Heights.	Number of Blades per Row.	Nozzle Opening Area.
1	60″	1	2″	$2\frac{3}{4}''$	510	Full Admission
2		1	$2\frac{5}{8}''$	$3\frac{3}{8}''$	506	
3		1	$3\frac{1}{4}''$	4″	480	
4		1	$4\frac{1}{8}''$	$4\frac{7}{8}''$	400	
5		1	$5\frac{1}{8}''$	$5\frac{7}{8}''$	365	
6		1	$6\frac{1}{4}''$	7″	300	
7		1	$7\frac{5}{8}''$	$8\frac{3}{8}''$	280	
8		1	$8\frac{1}{4}''$	9″	256	

Note.—If the revolutions of the L.P. turbine are less than that of the H.P. turbine, the L.P. blade heights will require to be increased to allow of equal steam flow per minute.

Dummy Rings.

The dummy ring packing of the original Parsons' design is now found, in many cases, to be unnecessary, owing to the reduced size of rotor and the efficient application of the new type of thrust known as the "Michell," which is fitted both on the turbine shaft to take up steam thrust and on the main or driving shaft to take up propeller thrust. It should be noted that with gearing, the propeller thrust is not taken up by the turbines, the latter having only the steam thrust, which, of course, requires to be taken up in some way (by balance pipes, etc.). The propeller thrust is transmitted through the driving shaft lengths to the Michell thrust block, which is usually fitted aft of the large gear wheel.

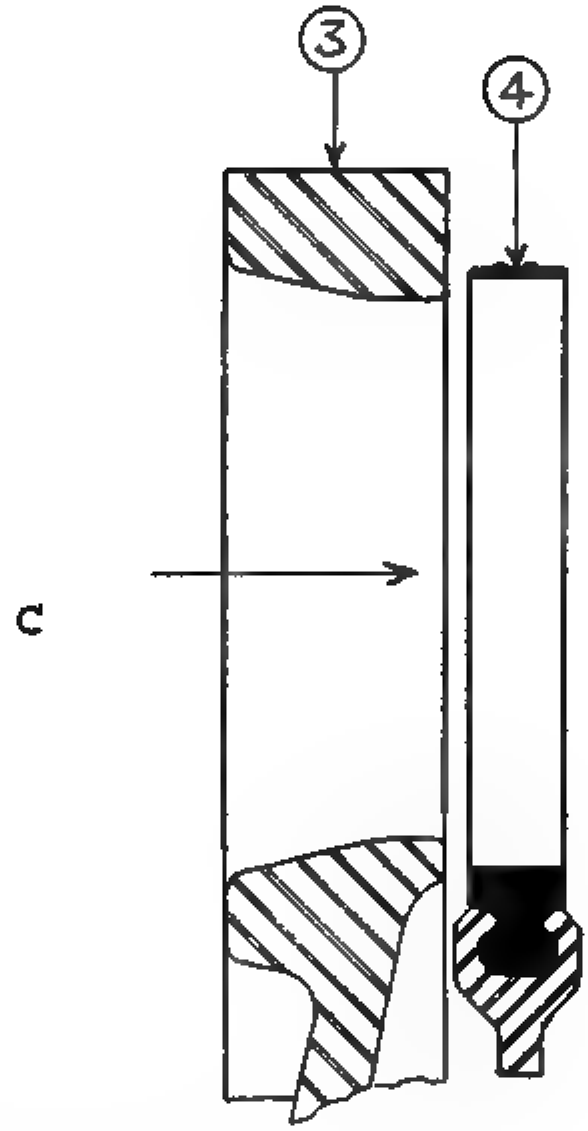
Astern Turbines (No. 4).

These, when fitted on the L.P. ahead spindle, often consists of a single velocity stage wheel, fitted with three moving blade rows and two fixed blade rows, the wheel being large in diameter, and the blades of maximum length. This arrangement allows of high power in small compass, although the efficiency is not of the best, the blade

No. 9.—H.P. Nozzle Plate of Impulse Turbine.

1. Nozzle plate. 2. First blade row.
3. Intermediate stage nozzle opening.
4. Blade row of intermediate stage.

friction loss and exit velocity loss being high. Blade tip leakage also acts detrimentally on the efficiency, and this leakage is due to the great difference in pressure between the initial steam of the stage, say, 126 lbs. guage, and the condenser pressure, say, 1 lb. absolute, and which allows of a direct blow through.

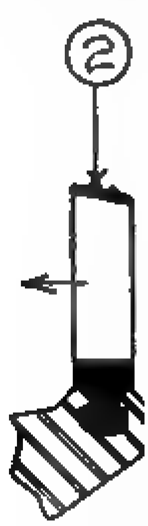


No. 10.—L.P. Nozzles of Impulse Turbine.

1. Nozzle plate of first stage.
2. Impulse blade of first stage.
3. Nozzle diaphragm of intermediate stage.
4. Impulse blade of intermediate stage.

The clearance C varies from $\frac{1}{4}''$ to $\frac{3}{8}''$, and the finer it is kept the less "spreading" action of the steam jet takes place.

The arrows show the direction of steam flow.



No. 11.—Astern Nozzle Plate of Impulse Turbine.

1. Nozzle plate.
2. Blade of first velocity stage row.

Operation of Curtis Turbines and Alquist Reduction Gears.

Referring to American standard ships, the following reprint from the *General Electric Review*, for January 1917, will be found of interest:—

"An interesting commercial feature of the present abnormal demand and production of steel cargo vessels in this country is the uniformity in capacity, speed, and power requirements. This has made it possible for the shipyards to standardise in the matter of hull construction and power plant requirements. Fully 90 per cent. of the turbine ships completed or under construction by the Pacific coast yards are provided with turbines of the same type and practically of the same size, namely, 2400 to 2600 H.P. A description of one may therefore be held to apply to all.

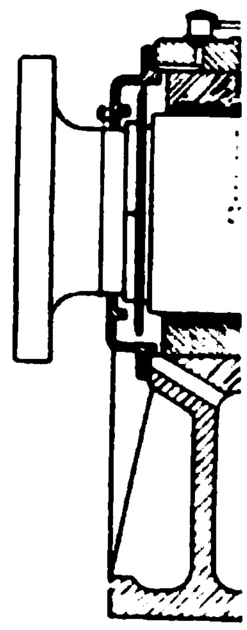
"The ahead turbine is designed to run at 3380 r.p.m. It is of the Curtis type of impulse turbine as developed by the General Electric Company, and consists of five stages, the first having two rows of buckets mounted on a single wheel and each of the succeeding stages a single row of buckets. The speed of the turbine is controlled by means of a lever operated balanced throttle valve in the main steam line, but in order to overcome the loss in efficiency due to throttling when running at reduced speed, two hand valves are provided which block off a number of the first stage nozzle sections. By this means it is possible to obtain 58, 75, 83, and 100 per cent. of full power with full steam pressure at the nozzles, resulting in a net saving of 3 to 5 per cent. in fuel when it is necessary to run the ship at reduced speed in a rough sea.

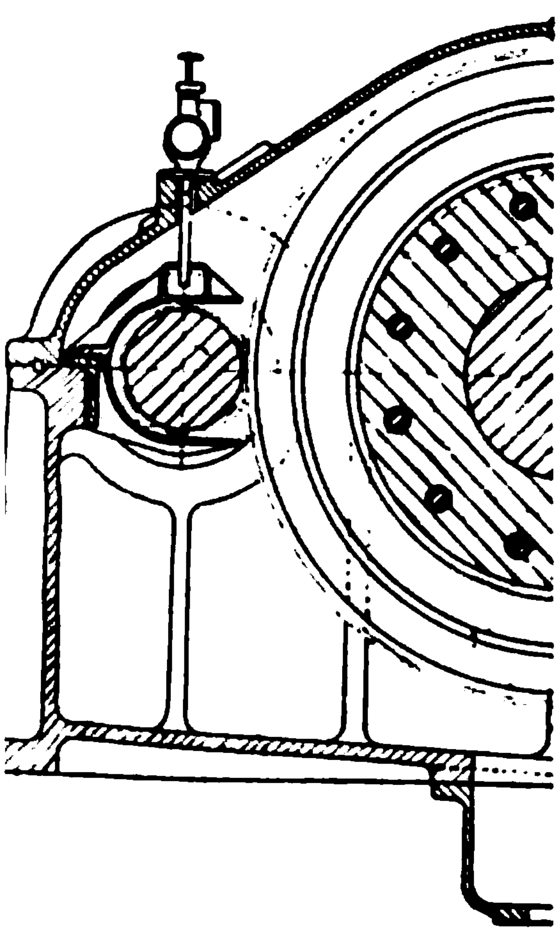
"The astern turbine has two stages of similar construction but of smaller diameter, and is mounted on the same shaft and in the same casing as the ahead turbine, both having a common exhaust. Steam is admitted by a balanced valve operated by the same lever controlling the admission valve of the ahead turbine. When the ship is running forward the wheels of the astern turbine revolve in a vacuum and therefore consume but little power.

"The reversing turbine will develop two-thirds torque at two-thirds speed with a total steam flow equal to that of the forward turbine at rated capacity. The nozzle area, however, is greater than that of the ahead turbine, thus permitting the reversing power to be considerably increased if desired.

"The speed of the propeller being 90 r.p.m. and the turbine 3380 it is necessary to obtain a gear reduction of 37·5. This is accomplished by means of a double reduction, the high speed gearing having a ratio of 7·36 and the low speed 5·10.

"The reduction gear is of the 'one-plane' type, that is, the axes of the high and low speed pinion and gear shafts lie in the same horizontal plane. This arrangement reduces the head room, simpli-





No. 12-

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fies lubrication, and facilitates inspection and accessibility to all parts.

"Power is transmitted through the high speed or driving pinion to two gears, one on each side, and thence through the two low speed pinions to the low speed gear. Rigid bearings are used throughout for the gears and pinions. This division of power between two low speed pinions possesses several obvious advantages, such as saving in longitudinal space required, weight of parts, width of gear teeth, and size of bearings.

"The turbine shaft is connected to the high speed pinion shaft by means of a slip coupling which prevents any part of the propeller thrust being transmitted to the turbine. The high speed gears and pinion and low speed pinions are therefore free to adjust themselves to the position of the low speed gear which is coupled solidly to the thrust shaft. The position of the turbine wheels with respect to their nozzles is fixed by a small adjustable thrust bearing on the turbine shaft.

"It is highly desirable on the score of simplicity to be able to use the same grade of oil for both the turbine bearings and the reduction gear, thus preventing duplication of oil pumps, strainers, coolers, storage tanks, settling tanks, etc. It has been possible to accomplish this by the use of a moderate tooth angle and by reason of the flexible disc construction of the gears. Several grades of oil were tried out on the S.S. 'Davanger,' the best results being obtained with a medium heavy oil having a viscosity of 260 (Saybolt) at 100° F. This oil proved to be about right for the dual purpose for which it was used, being light enough to flow freely through the turbine bearings and heavy enough to give sufficient cushioning effect at the face of the gear teeth. The smooth and quiet running of the 'Davanger' gears was remarked upon repeatedly by engineers and others present during the trial trip.

"Oil for the turbine bearings and gears is circulated by means of steam-driven pumps, which take the oil from the main tank and force it first through a strainer and then through a cooler before it passes through the spray nozzles supplying oil to the gears, or is delivered to the turbine bearings and the various bearings of the gears and pinions. A settling tank is also provided for removing any water which may get into the oiling system.

Economy.

"The engine room logs of the Union Oil Company's tank steamers 'La Brea' and 'Los Angeles' afford an unusual and valuable comparison in fuel consumption between reciprocating engines and Curtis turbines with reduction gears. These are sister ships both built by the Union Ironworks Company in 1916, and both operating in the same character of service and under similar conditions. The ships

differ from each other only in character of the propelling machinery and the cargo pumping systems. 'Los Angeles' is driven by a triple-expansion engine with a propeller speed of 65 r.p.m., and has the usual type of centralised steam pumping plant for discharging her cargo of oil. The propelling machinery of 'La Brea' consists of a 2600 B.H.P. Curtis turbine with Alquist reduction gears giving 90 revs. at the propeller. She is also fitted with a unique and highly efficient pumping system originated by Mr O. B. Kibele, Superintendent of Transportation of the Union Oil Company, in which a separate electric motor driven pump is provided for each compartment, power being supplied to the motors from a 300 Kw., 60-cycle, alternating current Curtis turbo-generator located in the engine room.

"Some of the important dimensions and data applying to the hull and propelling machinery of 'La Brea' are as follows:—

Hull.

Length over all	442 ft.
Length between perpendiculars	435 ft.
Breadth, extreme	56 ft.
Depth, moulded	33 ft. 6 in.
Dead weight carrying capacity	10335 tons.
On mean draught of	27 ft.
Oil tank capacity	70000 barrels.
Speed with 10000 tons dead weight	11 knots.

Propeller.

Diameter	16 ft. 6 in.
Pitch	14 ft.
Revolutions per minute	90.

Power Plant.

Turbine rating	2600 H.P.
Number boilers (Scotch type)	3.
Heating surface, boilers	2685 □'
Working pressure	210 lb. gauge.
Superheat	50° F.
Fuel oil—mechanical burners	...
Condenser cooling surface	4000 □'
Circulating water, gallons per minute	4000.

"It will be seen from the results shown that the increase in fuel consumption of the reciprocating engined ship 'Los Angeles' over

that of the steam-turbined ship 'La Brea' under operating conditions as found is :—

While steaming - - - - -	17·9 per cent.
While steaming and in port - - -	21·1 per cent.

No. 13.—Plan of Double Reduction Gearing, S.S. "La Brea."

(General Electric Co. of America.)

A single turbine (on right) gears by its pinion with two intermediate gear wheels, which again, by means of pinions, gear into the large wheel of the propeller shaft (on left).

The total gear-down ratio is as 40 : 1. The first reduction or intermediate gear wheels and pinions are sometimes termed "idlers."

The gear is of the Alquist type.

"The above comparison does not take into account the fact that the average speed of 'La Brea' is one-half knot better than 'Los Angeles.' If the speeds had been equal in each case, the advantage in favour of 'La Brea' would have been not less than 20 per cent. while steaming and 23 per cent. total while steaming and in port.

"When we consider that the engine of S.S. 'Los Angeles' was specially designed for high economy, the performance of 'La Brea' is seen to be such as to definitely establish the superiority of the geared turbine drive. With further improvements in the way of increased steam pressures, higher superheats and power plant design and equipment, there are still further possibilities in reduced operating costs of steamships driven by geared turbines which may be confidently expected in the future."

The following is a brief description of the auxiliary plant and the oil cargo pumping plant on the S.S. "La Brea."

The installation consists of:—

One 375 Kw. and one 125 Kw., 3600 r.p.m. three-phase, 60-cycle 240-volt generators, each direct coupled to Curtis condensing turbines with direct connected exciters.

The turbine exhausts into the auxiliary condenser, but is also arranged so that it can exhaust into the main condenser. When discharging, the larger turbine is of sufficient size to operate nine pumps at 140 lbs. pressure, but the necessary arrangements are made so that the two generators can operate in parallel, making a total capacity of 500 Kw.

Cargo Pumping.

This installation consists of:—

Twenty-two 4-in. rotary pumps, specially designed for handling highly viscous oils, etc., etc., and are also fitted with a steam jacket for handling asphaltum.

Each pump can deliver 350 gals. per minute against a total head of 350 ft. at 200 r.p.m.

The pumps are secured in the bottom of each compartment of the vessel, and are driven by 40 H.P., three-phase, 60-cycle, 220-volt motors.

In addition to the cargo pumps, the following motors are installed in the engine room:—

One 60 H.P., 720 revs., three-phase, 60-cycle, 220-volt motor direct connected to the circulating pump for the main condenser.

One 50 H.P., 3600 r.p.m. three-phase, 60-cycle, 220-volt motor direct connected to the boiler feed centrifugal pump.

One 35 H.P, 1800 r.p.m., three-phase, 60-cycle, 220 volt motor, direct connected to the centrifugal ballast pump.

U.S.N. Collier "Jupiter."

The following data, referring to the U.S.N. collier "Jupiter," is reprinted from the pages of *The Transactions of the American Society*

'al Architects, as given in a paper by Lieut. Robinson of the avy :—

ɜ machinery of this vessel is of the turbo-electric drive type, as designed and constructed by the General Electric Co. of ca, who are represented in Britain by Messrs The British on-Thomson Co. Ltd., Rugby.

—

ɜngth over all	542′ 2½″
eam, extreme	65′ 3″
'raught, mean designed	27′ 8½″
o-efficient of fineness, block	·7215

ıere are three double-ended, return, tubular, marine type Scotch s, and one upright donkey boiler. The main boilers have the following mensions :—

Length, over all	23′ ¾″
Diameter	16′ 3″
Diameter of furnace (inside)	3′ 4″
Number of furnaces per boiler	8
Length of grates	5′ 7½″
Grate surface per boiler	150 □′
Heating surface per boiler	6460 □′

'peller—

Diameter	15′ 11 $\frac{7}{16}$″
Pitch	14′ 5 $\frac{1}{16}$″
Expanded area	70·5 □′
Projected area	60·53 □′
Disc area	198·97 □′
Weight, one propeller	13900 lbs.

ırbo-Generator—

Weight of bedplate	16·3 tons
Length of bedplate	25′ 2⅛″
Length over all	27′ 7¼″
Height, extreme	9′ 5¼″

'urbine (Curtis Impulse Type)—

Weight	28·7
R.P.M. normal	1990
Number of stages	9
Number of rows of moving blades (two rows in first stage)	10
Number of rows of fixed blades (first stage)	1
Minimum clearance	·06″
Maximum clearance	·18″

Generator—

Weight	35·7 tons
Number of poles	2
Normal volts	2300
Normal amperes	1370
Normal frequency	33·2
Normal excitation	250 amperes
Normal r.p.m.	1990
Rated capacity	5450 K.V.A.
Rotor core length	60″
Air gap	$1\frac{3}{16}$″

Motors—

Number	2
Weight (two)	76·8 tons
Type (induction definite wound rotor)	"M"
Number of poles	36
Rated capacity	2750 H.P
Normal revs. per min.	110
Air gap	·1″

Trial Data—	**48 Hours**
Date of trial	Feb. 14, 1914
Speed in knots	14·99
Draught, mean	27′-9″
Displacement, in tons	19452

Pressure (gauge)—

Main steam, boilers, lbs.	193·8
Engine room, lbs.	185·8
Turbine, lbs.	168·4
First stage, lbs.	56
Forced lubrication, lbs.	30
Oil to governor relay, lbs.	80
Steam seals, lbs.	5
Auxiliary exhaust (absolute)	23·19
Air pressure to boilers, in inches of water	·712″
Vacuum, inches	28·2″

Revolutions of Double Strokes per Minute—

Starboard motor	116·72
Port motor	116·72
Main turbine	2·130
Wet air pumps	3·600
Circulating pumps	193
Dry air pumps	112
Feed pumps	22
Blowers	329

Number of propellers	- -	1
,, boilers	- - -	3
Boiler pressure (gauge)	- -	200 lbs.
Superheat	- - - -	200° F.
(Robinson superheater.)		

American Standard Ships—*continued.*

Vacuum - - - -	28·5″ (with 30″ barometer).
Turbines and generator revs. -	3600
Number of generators - -	2
Kilowatt output (each) - -	1250
Volts - - - -	2400
Phases - - - - -	3
Frequency - - - - -	60 cycles per second.
Number of motors - - -	2
Motor revs. - - - -	500 } Geared-down.
Shaft revs. - - - -	90 } Geared-down.

Auxiliary Gear—

All auxiliaries, such as pumps, fans, etc., are operated electrically by alternating current three-phase motors, and consist of the following:—

2 Condensate pumps - -	5 H.P.
2 Main circulating pumps - -	25 "
2 Lubricating oil pumps - -	15 "
1 Fresh water pump - - -	25 "
2 Ballast pumps - - -	5 "
1 Refrigerating motor - -	5 "
1 Workshop motor - - -	5 "
1 Anchor windless - - -	35 "
3 Deck winches - - -	25 "
8 Deck winches - - -	15 "
1 Steering gear motor - -	25 "
1 Forced draught motor fan - -	25 "

Gear-Down Ratio—

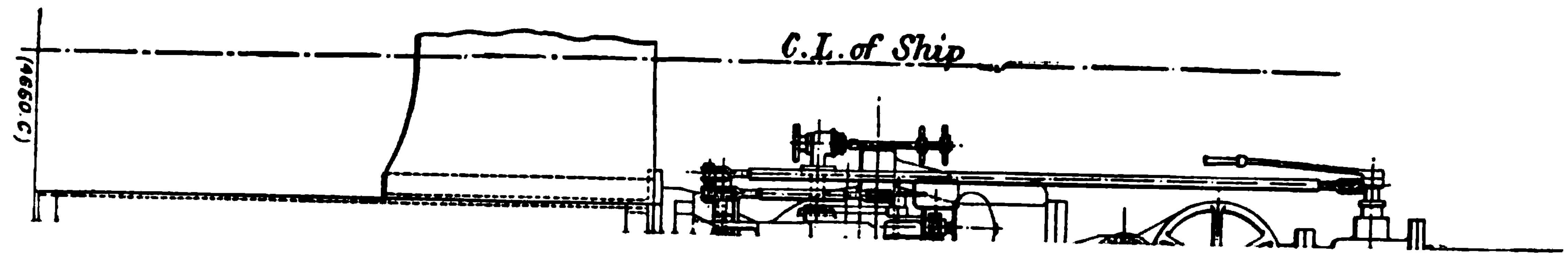
On looking over the above interesting data it will be noticed that the first speed reduction is between the generator and motor, from 3600 revs. to 50, or as ratio of 7·2 is to 1, as $3600 \div 500 = 7{\cdot}2$; the second reduction is obtained by gearing between the motors and the shaft, the ratio being as 5·55 is to 1, so that the combined gear-down ratio is $5.55 \times 7{\cdot}2 = 39{\cdot}9$, or, say, 40 to 1.

Operation of Machinery—

The electrical connections are operated by a master controller of the railway type, which actuates the necessary contactors for carrying out all manœuvring operations, such as starting, stopping, reversing, or reducing speed.

Description of Parsons' Turbines T.S.S. "City of Canton."

The high pressure and low pressure turbines work in series normally, and drive through their respective pinions one gear wheel connected to the propeller shaft, but each turbine can take steam direct from the boiler, and each has an astern turbine working on the same pinion. Thus, so far as the propelling engines are concerned, there are the



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... should be as thorough as possible. For this purpose sprayer nozzles distributed at suitable intervals are arranged to spray the oil direct on to the engaging teeth, thereby providing a film of oil between the surfaces in contact. The bearings

dvantages of duplication against the contingency of one turbine reaking down. The turbines transmit their power through flexible ouplings to the pinion shafts, the pinions being arranged one on each de of the gear wheel. The main shaft of the gear wheel is provided th a solid flanged coupling which is directly connected to the thrust aft. The reduction ratio is 1 to 17. The longitudinal half sections the two turbines show that the astern high pressure turbine is ranged with an independent casing at the forward end of the ahead gh pressure turbine, being connected to the latter through a flanged upling. The low pressure astern-going turbine is incorporated in e exhaust end of the low pressure ahead-going turbine casing.

In order to deal effectively with the high pressure superheated am—125 degrees Fahr.—which the boilers are arranged to supply, turbines, which are of the Parsons type, have been designed on impulse reaction principle, with an impulse wheel forming the tial stages in the high pressure ahead and the high pressure astern-ng turbines. The low pressure turbines, both for ahead and ern-going, consist entirely of stages of reaction blading. The use superheated steam has also necessitated the use of materials not inarily used in the construction of marine turbine machinery. The bine casings, generally made throughout of cast iron, have been in s case partially made of cast steel. The steam admission end of high pressure ahead turbine casing has been made in this metal, ile the whole of the high pressure astern turbine casing is of cast el. The blading of the impulse wheels, as also the earlier stages of ction blading, have been made of copper material instead of the ıal brass mixture, the copper withstanding the action of the super-ated steam much better. The rotating portions of the turbines— bodies of the rotors and the spindles—have been made entirely m ingot steel forgings.

The gearing is of the double helical type, the shape, size, and pitch the teeth being made in accordance with what experience has proved be satisfactory. The large gear wheel is divided vertically, and ach section of the wheel, consisting of boss, sides, and rim, is cast in ne piece, flanges being provided on the boss for bolting them together. Near the rim a circumferential tie-piece of cast steel is provided with lange connections to the wheel sections. Heavy forged steel rims on which the teeth are cut are shrunk on and secured to the cast steel portions.

The distance apart of the two sections of the rim is sufficient to llow of central bearings being fitted to the pinion shafts, to preserve the lignment. The gear wheel shaft is of forged ingot steel, while the pinions and shafts are of nickel steel.

For the efficient and silent running of this gearing it is necessary that the means of lubrication should be as thorough as possible. For this purpose sprayer nozzles distributed at suitable intervals are arranged to spray the oil direct on to the engaging teeth, thereby providing a film of oil between the surfaces in contact. The bearings

for the turbine spindles, together with the bearings of the gear wheel and pinion shafts, are also arranged to work under a system of forced lubrication.

In the arrangement of machinery it was found possible to place the condenser at a low level, and at the same time quite close to the low pressure turbine; this assists the drainage problem, and also does away with the usual overhead exhaust connection. The condensers are of the Weir uniflux type, made suitable for 27½ in. of vacuum in ordinary service conditions with the temperature of sea water at 85 degrees. For circulating water through the main condenser, a pump of the centrifugal type is provided. This pump is driven by a single cylinder open engine, and the pump and engine have been supplied by Messrs Drysdale & Co., Ltd., Glasgow. An air pump of the dual type, supplied by Messrs G. & J. Weir, Ltd., is fitted to work in conjunction with the condenser. The air pump is of the merchant service type, having gun-metal pump barrels with cast iron bases and tops, fitted with gun-metal buckets, manganese bronze pump rods, steel piston rods, and Kinghorn valves in gun-metal seats.

Forced Lubrication Systems.

In geared down turbine practice the forced lubrication system is usually in duplicate, the main bearings system, with oil cooler complete, being independent of that for the gear cases.

This double arrangement is advisable to ensure that the oil for the main bearings is free from any metallic matter which may possibly be picked up from the gear wheel teeth, and which might result in serious heating up of the bearings if allowed to enter with the oil.

This condition is more likely to obtain when the turbines are first put into service, and before the gear wheels and pinions have had time to work up into good contact surface adjustment.

Oil Baffles.

It is important that the lubricating oil for the main bearings be kept out of the steam glands which are near the bearings, and for this purpose V baffles are fitted as shown in the sketch (No. 1). Sometimes in addition a centrifugal "oil thrower" is also fitted on the shaft next the steam gland, the action of which is to throw out any oil which finds its way along the shaft towards the gland from the bearing adjoining. It need hardly be pointed out that if oil finds its way into the turbine various troubles may arise, such as the following:—

1. Damage to nozzles, turbine blades, etc., and possible corrosion in turbine rotor or casing.

2. Danger of oil in boilers causing buckled plates, corrosion, priming, which latter would again likely produce blade stripping in the turbines.

Vacuum Breaker.

After steam is shut off, and with the usual high degree of vacuum carried in the condenser, the turbines often continue revolving for some appreciable time afterwards, the effect of which is to retard stopping and reversing of the turbines; to eliminate this, a cock is fitted to the condenser which, when opened, admits atmospheric air, and then "breaks" or destroys the vacuum and checks the turbine revolutions. The connection is thus termed a "vacuum breaker."

Data from Practice.

Pressures and Temperatures.

The average range of pressures and temperatures obtaining in geared down turbine practice run somewhat as follows:—

TYPE 1.—

Combined Impulse and Reaction Turbines with, say, 120° Superheat at Engine Room Stop Valve.

		Pressure.
Turbines.	Boiler steam	220 lbs. gauge.
	H.P. nozzle box	205 „
	H.P. 1st stage	110 „
	L.P. „ „	10 „
	L.P. last „	27″ vac.
	Condenser	28·5″ vac.

TYPE 2.—

Impulse Turbines (H.P. and L.P.) with Superheat (say 150°) at Engine Room Stop Valve.

		Pressure Gauge.
Turbines.	Boiler steam	225 lbs. gauge.
	H.P. nozzle Box	210 „
	H.P. 1st stage	115 „
	L.P. nozzle box	25 „
	L.P. 1st stage	16 „
	L.P. last „	28″ vac.
	Condenser	28·3″ vac.
	Bearing oil pump delivery	·45 lbs.
	Oil pressure at oil cooler	10 „
	„ temperature at cooler inlet	103°
	„ „ „ outlet	85°
	Pressure at gear case oil pump delivery	44 lbs.
	„ „ „ cooler	9 „
	Oil temperature at cooler inlet	102°
	„ „ „ outlet	78°
	Maximum oil pressure at bearings	5 lbs.
	„ temperature of bearings	130°. (H.P. Ford).
	Gear case temperature	93°
	Thrust temperature	96°

Pressure in Receivers, etc.

Gauge Readings (starboard)—

Boiler steam - - - - -	210 lbs. per □″
H.P. ahead steam - - - - -	83 ,,
H.P. ahead bye-pass - - -	47 ,,
H.P. 2nd expansion steam - - -	47 ,,
H.P. 4th expansion steam - - -	12 ,,
L.P. ahead steam - - - - -	$5\frac{1}{2}$ in. vac.
H.P. astern - - - -	25 in. vac.
H.P. astern bye-pass - - -	25 in. ,,
L.P. astern - - - - - -	$28\frac{1}{2}$ in. ,,
Main condenser (inner) - - -	$28\frac{1}{2}$ in. vac.
Main condenser (outer) - - -	27 in. ,,
Auxiliary steam - - - - -	210 lbs. per □″
Auxiliary exhaust - - - - -	5 ,,
Oil service - - - - - -	15 lbs. per □″
Engine-room oil supply - - -	25 ,,
H.P. dummy clearance - - -	$\frac{24}{1000} = \cdot 024$
L.P. dummy clearance - - -	$\frac{39}{1000} = \cdot 039$
Revs. per min. - - - - -	200

Separate H.P. astern and L.P. astern turbines are fitted, and the difference in absolute pressure will be noticed, the back pressures in each astern turbine casing working out as follows (assuming a 30 in. barometer).

NOTE.—"Fan resistance" pressure in H.P. astern,
$=30''-25''=5''$, and $5''\div 2=2\cdot 5$ lbs. absolute.

NOTE.—"Fan resistance" pressure in L.P. astern,
$=30''-28\cdot 5''=1\cdot 5''$, and $1\cdot 5''\div 2=\cdot 75$ lbs. absolute.

Temperatures.—The following temperatures represent average practice and refer to a twin geared-down turbine set of a passenger steamer, fitted with oil-fired water-tube boilers and Schmidt superheater tubes.

Sea Temp.	Hotwell Temp.	Feed Temp.	Circulating Discharge Temp.		Engine Room Temp.	Aft Boiler Room Temp.	Forward Boiler Room Temp.	Superheated Steam Temp.	Oil at Burner Temp.	Lubricating Oil Temp.	
			Port.	Star.						In.	Out.
68°	110°	208°	83°	83°	82°	96°	80°	480°	160°	96°	86°
74°	120°	210°	86°	84°	84°	102°	92°	540°	160°	106°	92°

Speed, Revolutions, Horse-Power, and Consumption.

In turbine practice the revolutions vary almost directly as the speed, or putting it the other way, the speed varies directly as the revolutions.

The horse-power varies directly in a ratio rather less than the cube of the revolutions, as the following results prove.

Consumption at Low and High Speeds.

(Geared-Down Reaction Turbines with Superheat of about 100° F.)

Speed in Knots.	Revs.	Boiler Steam (Gauge).	H.P. Turbine Steam.	L.P. Turbine Steam.	Auxiliary Exhaust.	Condenser Vacuum	Coal Per H.P. Hour.
10	74	210	42	20″ vac. (about 5 lbs. absolute)	7 lbs.	28″	2·5 lbs.
15	110	210	127	9″ vac. (about 10·5 lbs. absolute)	9 lbs.	28·5″	1·3 lbs.
20	145	210	208	9 lbs. (gauge)	8 lbs.	28·75	1·13 lbs.

On looking into the above record it will be observed that the consumption per horse-power at 10 knots is more than double that at 20 knots, which indicates the serious loss of efficiency due principally to blade tip leakage, which occurs when running at low turbine speeds even when geared down. This loss should be appreciably reduced with the advent of double reduction gearing and the fitting of impulse turbines in place of reaction or combined impulse and reaction turbines.

The consumption per S.H.P. per hour is greatest at low speeds, and becomes less as the speed increases, in some cases falling off to less than half. Therefore the consumption varies directly with the speed, but in a ratio much less than the cube of the knots. This brings out clearly the fact that at low turbine speeds the blade tip leakage losses result in much reduced efficiency and economy. The introduction of double reduction gearing down will reduce to some considerable extent this disadvantage, as even at low driving shaft revolutions (half or quarter speeds) the turbines will still be running at fairly high revolution speeds, assuming the gear down ratio to be somewhere in the neighbourhood of, say, 40 or 35 to 1.

Pressures in Turbine Casings (H.P. and L.P.).

The following record of average pressures at the different expansions are representative of a geared-down combined impulse and reaction turbine of moderate power (with superheated steam):—

Boiler Steam.	Steam at H.P. Turbine.	Steam at H.P. Nozzles (see open).	Steam at H.P. Impulse Belt.	Steam at H.P. 4th Expansion.	L.P. Turbine Steam.	Steam at L.P. 5th Expansion.	Condenser.	Auxiliary Exhaust Pressure.
184 lbs. (gauge)	180 lbs. (gauge)	165 lbs. (gauge)	113 lbs. (gauge)	38 lbs. (gauge)	5 lbs. (gauge)	18″ vac. (About 6 lbs. absolute)	27½″ vac. (About 1¼ lbs. absolute)	4½ lbs. (gauge)
184	182	170 (10 nozzles open out of 16)	91	30	1¾	20″ vac. About 5 lb. (absolute)	28″ vac. (About 1 lb. absolute)	6 lbs.

With ten of the nozzles open the pressure in the nozzle box of those not open (six in number) was a few pounds below that of the H.P. impulse belt, owing to the steam flowing back through the nozzle plate openings into the box (see No. 21B to face page 380).

Data from Practice.

Boiler steam - - - -	200 lbs. gauge.
1st H.P. turbine expansion - -	85 „ „
2nd „ „ „ - -	57 „ „
4th „ „ „ - -	15 „ „
Closed exhaust pressure - -	7 „ „
Condenser vacuum - - -	28 in.
1st L.P. turbine expansion pressure -	2 in. vac. (14 lbs. absolute).
Pressure in astern turbine - -	25 in. vac. (2½ lbs. absolute).
Shaft revs. - - - - -	214.
H.P. dummy clearance - -	$\frac{21}{1000}$ in. (·021 in.).
L.P. „ „ - -	$\frac{23}{1000}$ in. (·023 in.).

Pressures in Expansions—

In an impulse type H.P. turbine with, say, six expansions and an L.P. turbine with, say, eight expansions, the pressures work out as follows in average practice:—

Boiler steam, 210 lbs. gauge; H.P. control box, 195 lbs. gauge.

	Gauge Pressure in Expansions.					
	No. 1.	No. 2.	No. 3.	No. 4.	No. 5.	No. 6.
H.P. Turbine	Lbs. 115	Lbs. 92	Lbs. 74	Lbs. 56	Lbs. 42	Lbs. 35

	Gauge Pressures in Expansions.								
	No. 1.	No. 2.	No. 3.	No. 4.	No. 5.	No. 6.	No. 7	No. 8.	
L.P. Turbine	Lbs. 32	Lbs. 20	Lbs. 8½	Lbs. 1½	12″ vac. (9 lbs. abso.)	21″ vac. (4½ lbs. abso.)	25″ vac. (2½ lbs. abso.)	27″ vac. (1½ lbs. abso.)	Condenser 28½″ vac. (¾ lb. abso.)

NOTE.—Barometer, 30″.

Pressures in Ahead Nozzles, etc.—

When running at full power the pressure of the steam after passing through the 1st stage nozzles usually drops to rather less than ⅔ of the initial steam pressure. Previous to this, the pressure will have dropped between the boilers and nozzle box to the extent of from 10 to 20 lbs.

The following examples from practice illustrate the foregoing:—

Example.	Boiler Steam (Gauge).	Gauge Pressure in Nozzle Control Box (Receiver).	Gauge Pressure at 1st Impulse Stage.
	Lbs.	Lbs.	Lbs.
1	200	185	92
2	182	174	110
3	215	190	115
4	179	164	112
5	220	190	110

When the power is reduced the pressure will fall off proportionately with the number of nozzles closed down.

NOTE.—It should be noted that the pressure at the 1st impulse stage is also the initial pressure at the 1st reaction stage, as, throughout the impulse stage of any turbine, the pressure remains practically constant, although the velocity of the steam falls off.

Dummy Clearance when Heating up and Changing Speed, etc.

1. If, under working conditions, the rotor expands more longitudinally than the casing (which is the usual experience), the dummy clearance will be reduced.

2. If the casing expands more longitudinally than the rotor, the

dummy clearance will be increased. This sometimes occurs in heating up, as shown in the typical example taken from actual practice.

	P.L.P.	H.P.	S.L.P.
Cold - -	·018	·026	·025
Hot - -	·037	·030	·034
Running -	·021	·021	·025

It will be noticed that the dummy clearance increased when heating up, but came back again when under running conditions.

3. In reducing speed, or in stopping turbines, if the casing contracts more than the rotor, the dummy clearance will be reduced, but if the rotor contracts more than the casing, then the dummy clearance will increase. As the first-mentioned condition usually obtains in practice, it will be evident that great care must be exercised in adjusting the dummy clearance to suit the various conditions of steaming; it is therefore advisable to run with a margin of safety clearance in the dummies, say about $\frac{35}{1000}$ or ·035, which setting should allow easily of all changes in speed variation, in stopping, or in going astern. In reversing, the ahead turbines run in a vacuum at low temperature, and the casings are at the limit of longitudinal contraction, thus bringing the dummy clearance down to a minimum.

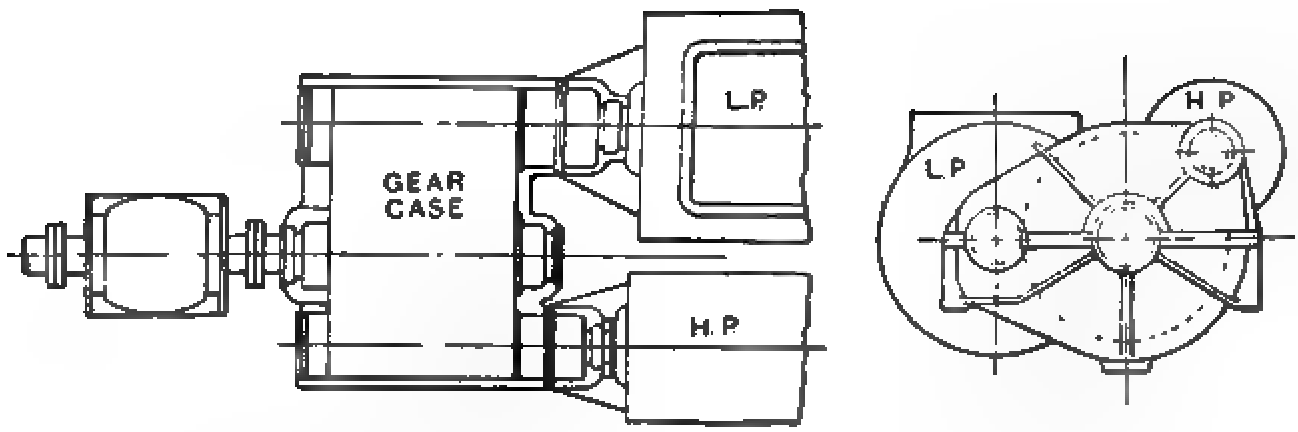
Dummy Clearance at Low Speeds.

When running at low speeds the after end of the turbine casings is much lower in temperature than when running at full speed, and the contraction resulting from this fall of temperature in most cases reduces the dummy clearance to within danger limits. For this reason it is in some cases advisable to keep the heating steam connection partly open when running at reduced speeds, so that the casing expansion may remain (as far as possible) constant, and thus keep the dummy clearance normal.

The dummy clearance can be regulated more or less by the amount of heating steam admitted, and by the time the heating connection is kept open.

As regards dummy clearance setting in general, it should be stated that hardly any two sets of turbines work out alike, so that no hard and fast rule can be laid down: each case must be experimented with individually to obtain reliable information as to the amount of clearance required under different conditions of temperature, speed, and power. In addition to this, it is now well known that the difference in economy obtained between running with, say, $\frac{10}{1000}''$ dummy clearance and $\frac{40}{1000}''$ dummy clearance is inappreciable; it is therefore not worth the risk involved to run the turbines with very fine dummy clearance settings.

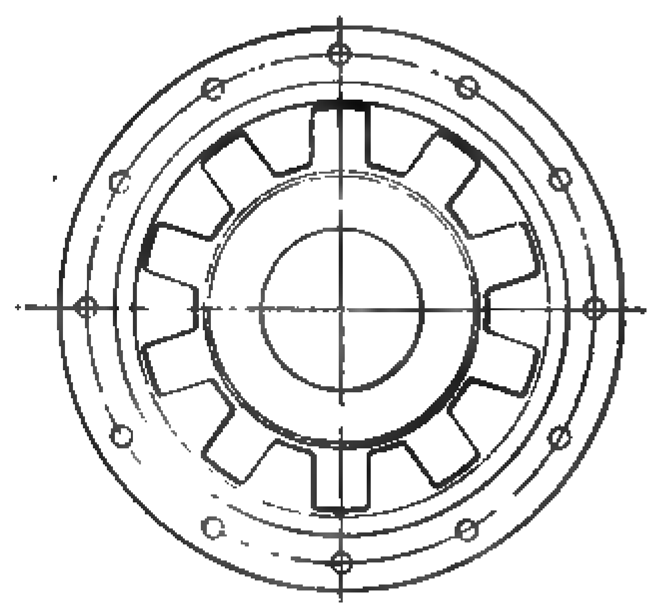
Arrangement of Turbines.



No. 25.*—Plan and End Elevation, showing Turbines (two), Flexible Couplings, Gear Case, and Michell Thrust Block (on left).

The centre of the H.P. turbine is placed higher in position than that of the L.P. turbine, which arrangement results in the saving of space, and of the turbines being placed closer to each other. This also reduces the lengths of exhaust piping, etc., required to connect up the two turbines.

Flexible Coupling.



No. 26.*—Flexible Coupling.

The coupling is made up of two separate steel toothed claw pieces A and B, each centred and keyed on its respective length of turbine spindle and pinion shaft, and the whole is held in place by thin wrought iron nuts. The sleeve C is made in two parts, with a centre flanged joint, and secured by fitted bolts.

On the inner circumference teeth are formed, which mesh with those of A and B, thus giving a positive connection between rotor and gearing, but at the same time one in which slight non-alignment between the shafts is taken up, by reason of the periphery of the teeth on A and B being, as it were, part of a sphere.

Lateral expansion of the rotor shaft is taken up by the clearances allowed in a longitudinal direction.

In order that the working faces of the teeth may be lubricated, a special oil-scoop is fitted to catch the oil flowing from the bearing end, and transmit it to the teeth.

The teeth are radial, with corners well rounded off, and adequate clearance at the tips to allow of the maximum adjustment being made without locking of teeth.

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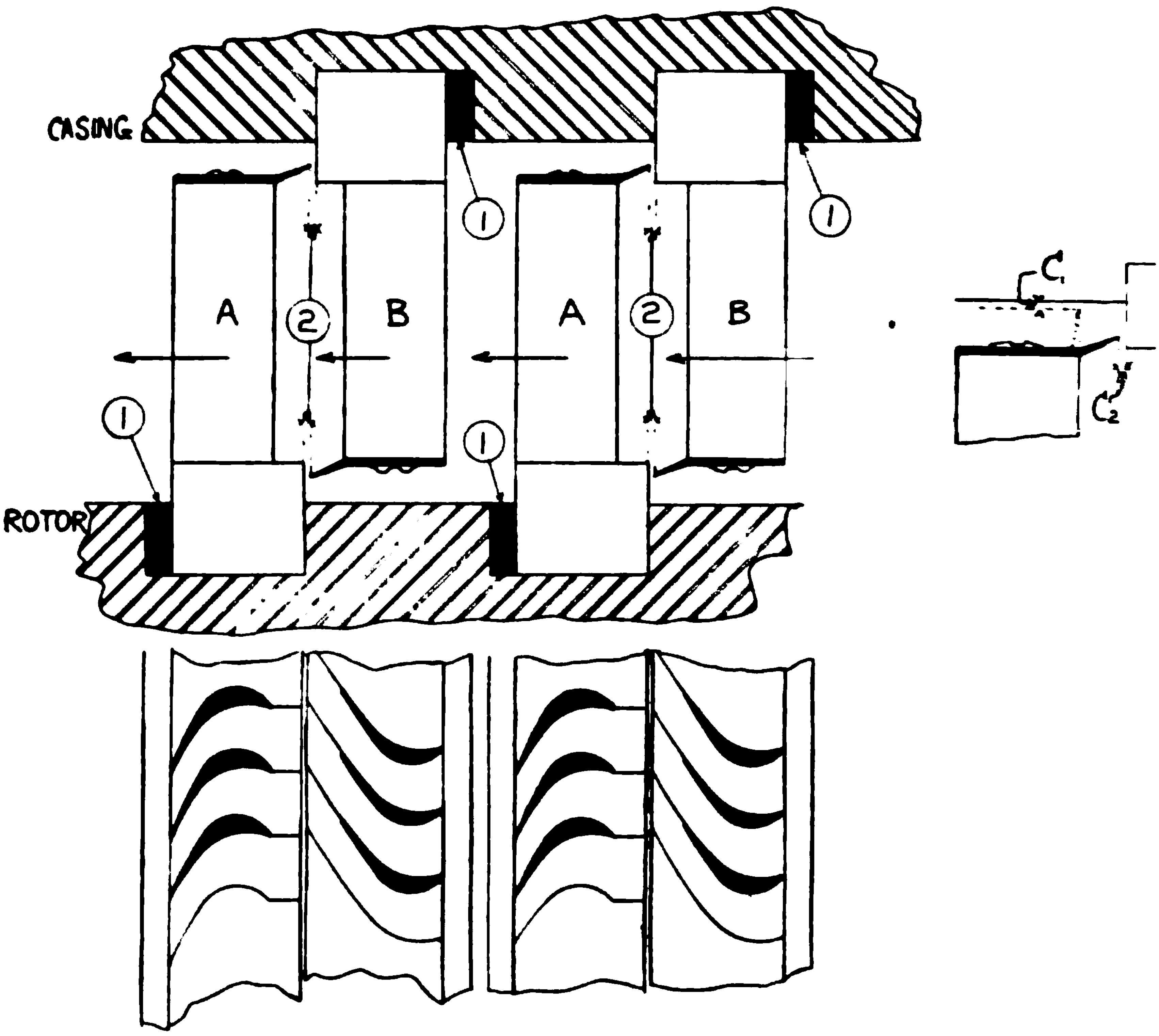
Blade Tip Leakage.



No. 27.—Longitudinal Method of Reducing Blade Tip Clearance.
(Parsons' Turbines.)

A, A. Rotor blades.
B, B. Casing blades.
1. Double caulking strips or packers.
2. Longitudinal clearance of from ·015″ to ·020″.

The small sketch on right shows the usual method (dotted lines) of shroud ring with the radial fin or knife edge, which gives a radial clearance of, say, ·025 or ·030 C_1, whereas C_2 shows the clearance allowed for in a longitudinal direction by bending over the knife edge through nearly a right angle, as arranged in the newer design.

The practical objection to this arrangement lies in the difficulty of allowing accurately for longitudinal change of position in the rotor due to expansion, contraction, or alteration of dummy setting.

Double Wire Drawing Rings.

No. 28.—Dummy Rings.

The type of Parsons' dummy ring is two-fold in its action, as the wire-drawing effect is carried out both longitudinally and radially, as shown clearly by the arrows.

Brady and Martin's Kenotometer (Vacuum Gauge).

Description.—Consists of a chemically cleansed mercury tube A, fitted with two air traps in the large bulb B, and a capillary constriction so as to make the instrument safe from damage during vibration, etc.

No. 29.

It is impossible for mercury to find its way into the condenser owing to the position of the traps in bulbs B and C.

At the side of trap C is placed an improved type of drying arrangement, D, consisting of a stout glass tube with brass mounts at each end and brass connecting tubes with brazed joints. At the lower end a screw cap is provided which can be loosened by means of an ordinary spanner. If it becomes necessary to replace the drying material (which only happens at long intervals), the cap is unscrewed and the glass sleeve containing the desiccating material can be withdrawn, refilled, and replaced without disturbing any of the other fittings.

The limb C is connected to the air inlet valve E, which when opened allows air to flow in before disconnection from the condenser: if, however, this connection breaks accidentally, even then no damage to the instrument would result, owing to the capillary constriction portion of tube A.

K is a water check valve which automatically prevents water being drawn into the gauge if the condenser vacuum disappears altogether. Two graduated opal scales are fitted moveable by rack and pinion from outside of the case, and these allow of very accurate adjustment for the difference of mercury level of the two limbs.

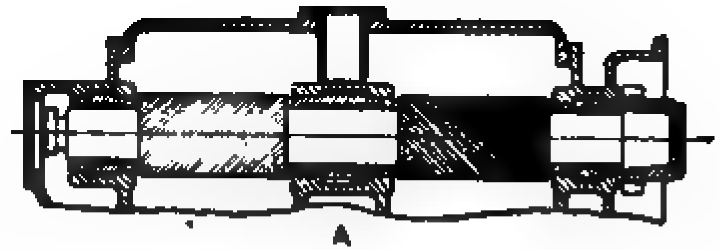
One scale (in black) is graduated in inches divided into tenths, the other (in red) shows its absolute pressure in the condenser, together with the corresponding percentages of a perfect vacuum (in black), and the latter scale is graduated on the basis of the standard barometric height of 30 in.

A barometer difference of 1 in. above or below this would, with the high vacuum usually carried in turbine practice, give a scale error less than the observation error, and therefore become negligible.

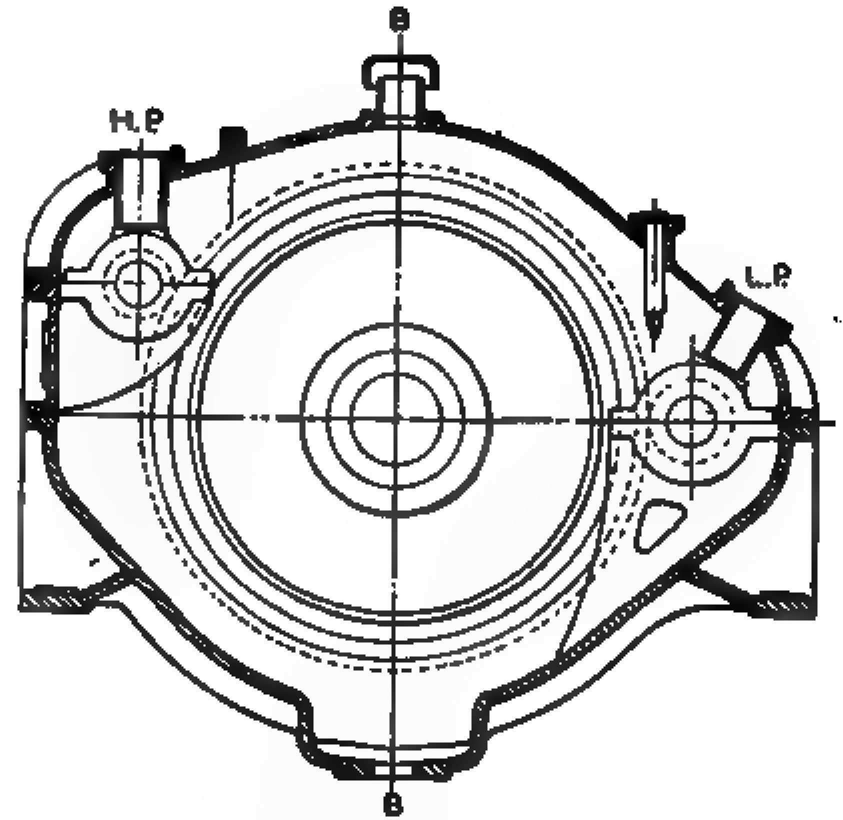
To Read the Kenotometer.

The scale is moved by the rack and pinion until the *red arrow line* is level with the mercury in wide limb B of the U tube; the height of the mercury in the narrow limb A is then read off.

For 1 in. difference in height of the barometer from the 30 in. standard the error is only $\frac{1}{30}$ of 1 per cent. (or ·033 per cent.) for each percentage division.



SECTION AT B-B.



SECTION AT A-A.

No. 30.*—Sectional Views of Gear Wheel, Pinions, and Gear Case.

It will be observed that in the type illustrated the pinions are not placed diametrically opposite on the gear wheel, but for reasons of space saving are arranged nearer to each other as shown.

The pinions are of the three-bearing type, and the large gear wheel has two bearings only.

The hand feeler pipes are shown at the positions marked H.P. and L.P., the vapour escape pipe and hood at B, and one of the oil jet nozzles directed on the teeth meshing contact of the L.P. turbine. The direction of shaft rotation is therefore clockwise.

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DESCRIPTION OF PROPELLING MACHINERY FOR S.S. "AURANIA."

(*See Frontispiece.*)

THE propelling engines consist of two sets of turbines, each set driving a propeller shaft through mechanical gearing, the vessel being of the twin screw type.

It is contemplated that the power developed by the turbines will, in the aggregate, correspond to about 7200 shaft H.P. in ordinary service at sea, the revolutions of the propeller under these conditions being about 100 per minute.

The machinery has been constructed under the survey of the Board of Trade for a passenger certificate, it also complies with the Rules and Regulations for Lloyds' classification, and in addition will pass the American Survey.

Passing on to a more detailed description of the various units in the boiler rooms and engine room, reference, first of all, will be made to the boilers, which are four in number, arranged in two boiler rooms.

The boilers are of the multitubular, double-ended, return tube type, 17 ft. 6 in. mean diameter, and 20 ft. 9 in. over end plates. Each boiler is fitted with eight furnaces of the Morison suspension type, and with back ends suitable for easy withdrawal. A separate combustion chamber is provided for each furnace. The boilers have been constructed throughout so as to permit of a working pressure of 190 lbs. per square inch being maintained.

Superheating arrangements are fitted to all boilers, the special fittings for these, consisting of tube elements and headers, having been supplied by Messrs The North Eastern Marine Engineering Co. Ltd., Wallsend-on-Tyne.

All steam stop valves through which superheated steam passes have cast steel chests with steel fittings.

Diamond blowers have been arranged in connection with each combustion chamber, with all fittings complete.

The Brundrit patent temperature balance has been fitted to each boiler for promoting automatic circulation of the water.

The whole of the boiler installation has been designed for working under natural draught conditions, and the furnace fittings have been arranged in accordance with the Cunard Company's special requirements, embodying such details as firebars of the "Trident" pattern, and patent backbridges as supplied by Messrs Railton, Campbell & Crawford, Ltd.

Two chimneys are provided, of elliptical section, and having an outer casing about eighteen inches clear of the main chimney, the height of the chimneys beings about 130 ft. above the base line. The uptakes and chimneys are fitted with division plates for each

boiler, these being carried the full height of the chimneys so as to make the draught for each boiler independent. Sectional dampers are also provided in the uptakes for isolation purposes. At the forward side of the forward funnel is fitted a set of triple bell whistles in connection with a set of Willet-Bruce control gear, the whistles and gear having been supplied by Messrs T. Downie & Co. of Liverpool.

As previously mentioned, the vessel is arranged for twin-screw propulsion, the port and starboard lines of shafting being driven each by a set of geared turbine machinery. Each set consists of a high and low pressure turbine, working in series and coupled to the pinion shafts by flexible couplings, the pinions being situated one on each side of the gear wheel, which in turn is coupled direct to the propeller shafting.

The astern going turbines are incorporated in the exhaust end of the low pressure turbine casings.

The turbines are of the Parsons' design, and of the impulse reaction type, an impulse wheel being introduced, as the initial stage in the high pressure ahead and the high pressure astern turbines, to take the first drop in pressure. The impulse wheels are followed in each case by the usual reaction stages, while the low pressure turbines consist entirely of stages of reaction blading.

The gearing is of the double helical type, arranged with a spiral angle and depth of tooth, which experience has shown to be most suitable. The main gear wheel is constructed with a cast steel central portion to which a heavy steel forged rim is attached. The pinions are of nickel steel. Central bearings are provided for the pinion shafts to ensure the necessary alignment being preserved.

For the efficient and silent running of this gearing, it is essential that the lubrication arrangements should be as perfect as possible, and as a means to this end, sprayer nozzles distributed at suitable intervals are arranged to spray the oil direct on to the engaging teeth, thereby providing a film of oil between the surfaces in contact.

The shaft bearings for the turbines and the gearing are also arranged to work under a system of forced lubrication.

The rotating portions of the turbines, comprising the bodies of the rotors and the spindle ends, are made entirely from ingot steel forgings.

The turbine casings, which in an ordinary way would have been entirely of cast iron, have in this case been constructed partially in cast steel, in order to withstand the superheated steam with which it is intended to work these turbines. Those portions of the turbines which are made in cast steel are the steam admission ends of both the high pressure ahead and high pressure astern turbines; similarly, the material of the blading is partly of copper and partly of the usual brass mixture, the copper sections being fitted to the impulse wheels and the earlier stages of reaction blading.

The floor level at the starting platform is at about the centr line of the turbines, and the whole of the manœuvring valves ar supported on a pedestal, suitably placed, so that the control whee on the valve spindles are at a convenient height for operating; th pressure gauges are also grouped in close proximity, the who forming a compact and easily controlled arrangement. Pipe cor nections are taken from the valves to the steam strainers, which ar directly attached to the turbines.

The thrust shafts, main line shafting, and propelling shafts ar of forged ingot steel, the strength of the shafting being considerabl in excess of Board of Trade and Lloyds' requirements.

The propellers are of the built up type with four manganes bronze blades to each, which are secured to the propeller bosses b Lowmoor iron studs and gun metal cap nuts. The bosses are mad of cast steel.

The condensers are of the Weir "Uniflex" type, and are place at a level such that the low pressure turbines drain into them; i this way the accumulation of moisture at the low pressure end o the turbines is reduced to a minimum.

The above arrangement also obviates the use of the somewhat objectionable overhead connecting trunk between the low pressure exhaust and condenser.

Air pumps of the "Dual" type, as supplied by Messrs G. and J Weir, Ltd., are fitted to work in conjunction with the above con- densers. The condensing plant is made capable of maintaining a vacuum of 28½ in. when the machinery is working under the sea service conditions, with the barometer at 30 in. and sea water at 60° F.

Two centrifrugal circulating pumps are fitted for supplying the circulating water to the main condensers. These pumps are each fitted with one single cylinder, double-acting, enclosed forced lubri- cation engine, and have been supplied by Messrs M. Paul & Co. Ltd, Dumbarton.

The boiler feed pumps consist of two pairs of direct actin vertical pumps, each pair of pumps being capable of delivering th necessary feed water under ordinary service conditions.

The feed pumps deliver the feed water through one large surfa feed water heater of the multiflow type, which is designed to utili the exhaust steam from the various auxiliaries.

The feed pumps and feed heater are all of Messrs G. and Weir's well-known make.

The whole of the foregoing pumps have United States metal packing fitted to the piston rods.

A feed water filter of the "Cascade" type, as supplied by Mes The Contraflo Condenser and Kinetic Air Pump Co. Ltd., is fitt the filter tank being provided with the necessary float gear controlling the feed pumps.

The air pumps are arranged to deliver the feed water to t filter tank, and the feed pumps to draw therefrom.

An auxiliary condenser of Weir's uniflux design is fitted, and is capable of dealing with the exhaust steam from the whole of the auxiliary and deck machinery. This condenser is of the non-vacuum type, no air pump being required under these conditions.

Circulating water for this condenser is provided by a centrifugal pump driven by a single cylinder enclosed forced lubrication engine supplied by Messrs M. Paul & Co. Ltd.

The condenser drains to a special tank having automatic gear which controls a vertical duplex feed pump, both tank and pump having been supplied by Messrs Thom, Lamont & Co. This pump delivers the feed water from the tank to the boilers through a discharge filter supplied by Messrs Railton, Campbell & Crawford of Liverpool. Alternative suction connections from other pumps are also fitted to this tank.

In connection with the forced lubrication system there are fitted three single cylinder direct acting vertical pumps, which have been supplied by Messrs G. and J. Weir of Glasgow. These pumps are arranged with suctions to the oil drain tanks situated in the double bottom of the vessel, and arranged to deliver through an oil cooler of multitubular design.

The connections are such that delivery can be made either to gravitation tanks or direct to the gearing and turbine shaft bearings, the oil after use finally draining back to the tanks in the double bottom.

For supplying circulating water to the oil cooler, a duplex pump of the vertical type, as made by Messrs Thom, Lamont & Co., is provided and fitted with all the necessary connections.

Other auxiliary machinery fitted by the machinery contractors in the vessel comprises the following:—Sanitary, bilge, and fire pumps. Two of these, of the vertical duplex type, have been fitted with suctions from the sea and bilges, and arranged to deliver overboard, and on deck for sanitary, wash deck, and fire service purposes.

There are also a bilge and a sanitary pump worked from the end of the gear wheel shaft, through a crank pin and connecting rod. The pumps have similar connections to those named for the duplex pumps.

An additional bilge pump of the vertical duplex type has been installed in the forward boiler room, with connections to draw from the bilges and deliver overboard.

The ballast pump is of the vertical duplex type, capable of discharging 200 tons of water per hour. This pump can draw from any forward and deliver into any after tank. This pump has also connections for performing various other duties.

For dealing with the fresh water for washing and drinking water purposes, there are two pumps of the vertical duplex type, with suction and delivery connections as necessary.

The whole of the above duplex pumps have been supplied by Messrs Thom, Lamont & Co.

The evaporating and distilling plant includes the following:—Two evaporators, each being capable of producing 50 tons of water per twenty-four hours. The vapour from these evaporators can be taken by alternative connection to the condensers, low pressure turbines, feed heater, and distiller.

The distiller is capable of condensing 50 tons of washing and drinking water per twenty-four hours at 65° F., with circulating water at 60° F. The evaporators and distiller have been supplied by Messrs Caird & Rayner of London.

For providing the feed water for the evaporator a pump of the vertical single cylinder type is fitted, this pump having been supplied by Messrs G. and J. Weir, Ltd. of Glasgow.

A feed filter, supplied by Messrs Railton, Campbell & Crawford of Liverpool, is fitted in the delivery pipe line of fresh water from the distiller.

Circulating water for the distiller is supplied by a vertical duplex pump of Lamont's make.

The auxiliary machinery generally is made suitable for a working pressure of 190 lbs. per square inch, the air pumps and feed pumps are also suitable for working with a superheat of 50° F. The duplex pumps are capable of performing their respective duties with a working pressure of 150 lbs. per square inch, and all pumps are large enough for their work when exhausting against a back pressure of 20 lbs. per square inch.

The water ends of the pumps in most cases are made of gun metal.

The question of radiation of heat from external hot surfaces, extending from that of the boilers to exhaust and feed pipes, more particularly now that superheated steam is employed, has received careful attention, and all parts from which heat can escape have been covered with some form of non-conducting material.

A workshop has been fitted out in the discharge valve recess on the lower deck, and there has been installed a 6½ in. centre double-geared lathe having an 8 ft. gap bed, and is of an improved self-acting sliding, surfacing and screw-cutting type. There is also a 30 in. double-geared pillar drilling machine having a 2¼ in. spindle with 10 in. feed, either hand or automatic. In addition there is one 30 in. by 4 in. grindstone provided with fast and loose pulley.

The foregoing machines are driven by a 3 H.P. totally enclosed electric motor suitable for 112 volts direct current shunt wound. For transmitting the power from the motor to the machines, the necessary counter shafts and pulleys are fitted.

The whole of the above machines, together with the motor, have been supplied by Messrs C. Booth & Co., Liverpool.

For dealing with the ashes and other refuse that accumulates in the boiler rooms, there is provided in each of these compartments an 8 in. silent ash hoist, as made by Messrs Railton, Campbell & Crawford of Liverpool. Special shafts are constructed independent

of the ventilators for the above hoists. In addition, the stokehold ventilators are fitted with a small windlass and chain.

The ventilation of the engine and boiler rooms has been carried out in a very complete and effective manner, and for both compartments the ventilating trunks below the casing top are built as part of the vessel's structure.

For the engine room there are two trunks, each having a 48 in. diameter tube and cowl above the engine room casing, the cowl being turned by gearing from below. At the lower end of each trunk is fitted a 45 in. Keith open type centrifugal fan, provided with deflector.

The fan is driven by a vertical shaft enclosed electric motor of Messrs Laurence Scott & Co.'s make. Each of the above fans, when running at a speed of 600 r.p.m., is capable of dealing with 30,000 cub. ft. of air per minute.

There being four stokeholds, each one is provided with two ventilating trunks, which are surmounted with 48-in. diameter tubes and cowls, turned by gearing from a lower deck level. The space between the inner and outer chimneys is arranged as the means of egress for hot air from the boiler tops.

Alternative Types of Marine Machinery.

The following Tables I. and II., reprinted from the comprehensive and carefully compiled paper, "Some Alternative Types of Propelling Machinery for a 19½ Knot Steamer," written by Mr James Dornan, and read before the Meeting of the Institute of Engineers and Shipbuilders in Scotland, on 26th October 1915, should prove of interest to all students of marine engineering.

Table II. is of particular value, showing clearly, as it does, the advantages and drawbacks of one system of propulsion as compared with another.

It is only fair, however, to point out that Mr Dornan personally advocates the turbo-electric drive system.

The various designs are described by Mr Dornan as follows:—

A. Two sets of quadruple-expansion steam engines driving twin-screw propellers direct at 85 revs. per minute. Saturated steam is raised in six double-ended and four single-ended cylindrical boilers working at 210 lbs. pressure per square inch under Howden's system of forced draught. The engines are designed for a vacuum in the condensers of 27 in. at 55° Fahr. sea temperature and 30 in. barometer. All auxiliary steam is utilised in the feed-heater, where a feed temperature of 210° Fahr. is realised.

B. One set of either Parsons or Brown-Curtis turbines is arranged in series on four shafts running at 290 revs. per minute. The H.P. and I.P. turbines are on the outer shafts, and the two L.P.'s on the inner shafts. Reversing turbines are incorporated in the low-pressure ahead turbine casings. Each shaft has one direct driven

propeller. The boilers are as in A, with the working pressure reduced to 200 lbs. per square inch. The vacuum has been increased in this design to 28 in. under the same conditions as A, and the auxiliary steam is also employed in the feed heater.

C. This design is similar to A; only superheated steam is used in the engines. The resultant economy enables the work to be done in six double-ended and three single-ended boilers, *i.e.*, a single-ended boiler less than A. The superheater is of the Schmidt smoke-tube type, designed to maintain at the engine stop valve a superheat of 200° Fahr.

D. This is as B; only superheated steam is used. The boilers are the same as for C, with the pressure at 200 lbs. per square inch. 100° Fahr. superheat is used in this design.

E. and F. These designs are for geared turbines arranged on two shafts. The former consists of two Parsons or Brown-Curtis double-flow turbines running at 1000 revs. per minute, and driving twin-screw propellers at 200 revs. per minute through hydraulic gearing of the Föttinger type. The latter has four similar single-flow turbines arranged in series, driving the propeller shafts through mechanical gearing usually associated with the name of Parsons. The revolutions for F are 1800, and 160 per minute for the turbines and propellers respectively. The boilers are the same in the two designs, excepting the degree of superheat, E having 200° Fahr., and F 100° Fahr. superheat. The working pressure is 200 lbs. per square inch, and the vacuum under conditions as in A, 28½ in.

G. The two shafts of this design are driven at 85 revs. per minute through mechanical gearing of the Parsons type by electric motors running at 500 revs. per minute. These in turn are driven from two Ljungström turbo-dynamos which run at 3000 revs. per minute. As in E and F, the vacuum is 28½ in., with a working pressure of 200 lbs. per square inch; only a superheat of 260° Fahr. is maintained at the turbine stop valve by superheaters of the same type as fitted in C, D, E, and F proposals. The boilers are seven in number—four double-ended and three single-ended, all of the cylindrical type. The engine room casing of this design is closed, so as to prevent access of sea water down the hatch to the dynamos and motors, provision at the same time being made for ample ventilation of all spaces.

TABLE I.

COMPARISON OF HORSE POWERS, EFFICIENCIES, ETC.

	Length.	Breadth.	Depth.	Draught.	Displacement.	Speed.
1	600 feet	72 feet	46 feet	27 feet	21000 tons	19¼ knots

10000 NET REGISTER TONS.

	Reference Letter.	A	B	C	D	E	F	G
	Design.	2-Shaft Quadruple Engines.	4-Shaft Direct Turbines.	2-Shaft Quadruple Engines.	4-Shaft Direct Turbines.	2-Shaft Hydraulic Gears.	2-Shaft Mechanical Gears.	2-Shaft Turbo-Electric Gears.
2	No. of shafts	2	4	2	4	2	2	2
3	Revolutions per minute	85	290	85	290	200	160	85
4	E.H.P. naked	11430	11100	11430	11100	11430	11430	11430
5	E.H.P. with appendages	12500	12400	12500	12400	12300	12300	12500
6	Wake	0·165	0·20	0·165	0·20	0·165	0·165	0·165
7	Thrust Deduction	0·15	0·15	0·15	0·15	0·15	0·15	0·15
8	Efficiencies—Hull	0·99	1·02	0·99	1·02	0·99	0·99	0·99
9	Propeller, model	0·695	0·62	0·695	0·62	0·64	0·64	0·70
10	Propeller, actual	0·647	0·577	0·647	0·577	0·595	0·595	0·651
11	Mechanical	0·903	0·97	0·903	0·97	0·98	0·97	0·97
12	Propulsive	0·577	0·57	0·577	0·57	0.577	0·572	0·625
13	I.H.P.	21650	...	21650	...	...	...	...
14	S.H.P.	...	21800	...	21800	21300	21350	20000
15	Propulsive coefficient from E.H.P. naked	0·528	0·51	0·528	0·51	0·537	0·536	0·571

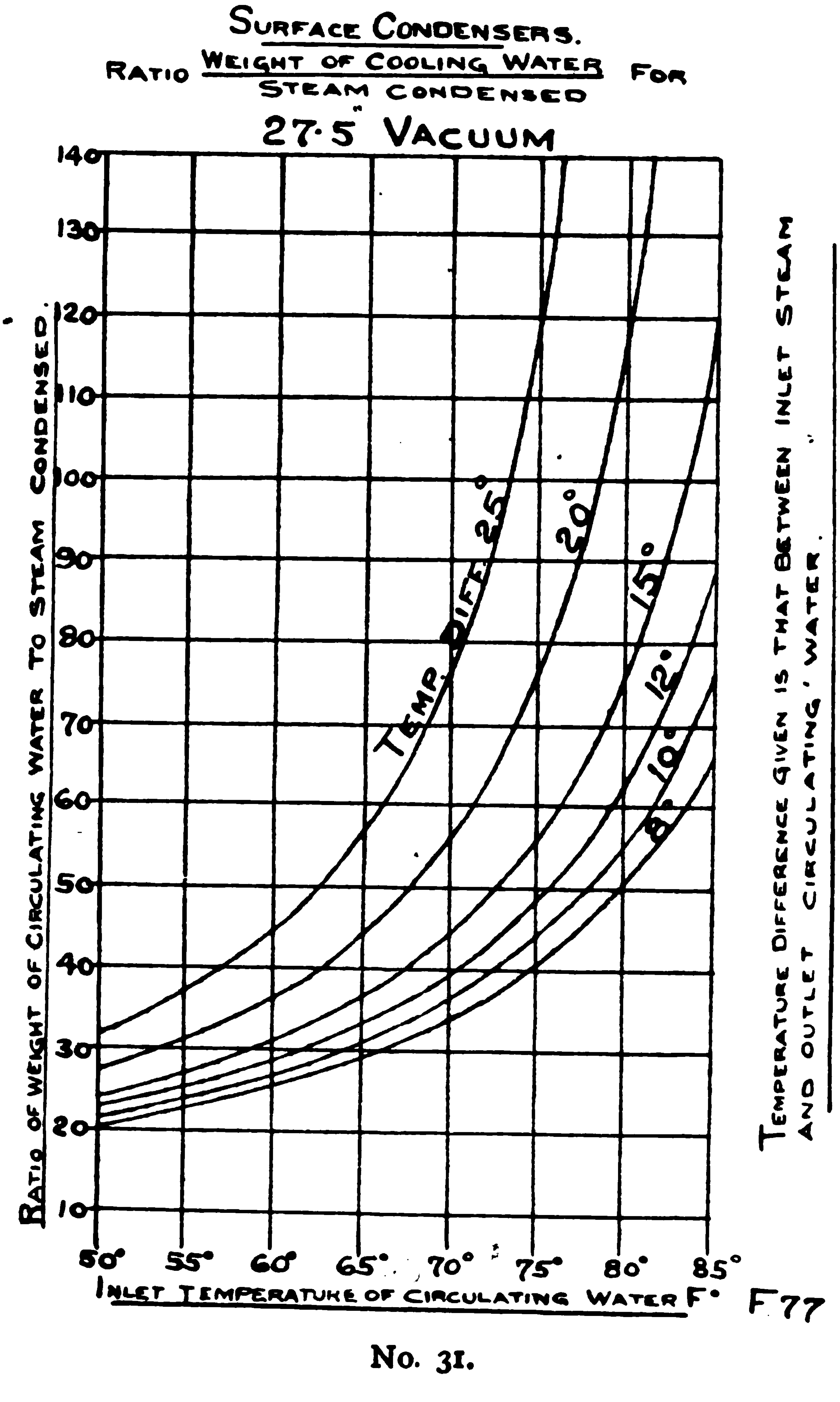
Table II.

Comparison of Coal Consumptions, Weights, Etc.

	Reference Letter.	A	B	C	D	E	F	G
	Design.	2-Shaft Quadruple Engines.	4-Shaft Series Direct Turbines.	2-Shaft Quadruple Engines.	4-Shaft Series Direct Turbines.	2-Shaft Turbines, Hydraulic Gear.	2-Shaft Turbines, Mechanical Gear.	2-Shaft Turbo-Electric Gear.
		Saturated Steam.		Superheated Steam.				
1	I.H.P.	21650	...	21650	...	...	...	...
2	S.H.P.	...	21800	...	21800	21300	21350	20000
3	Primary H.P.	21650	21800	21650	21800	23450	21800	22600
4	R.P.M.—Propellers	85	290	85	290	200	160	85
5	Turbines	...	290	...	290	1000	1800	3000
6	Working pressure, lbs.	210	200	210	200	200	200	200
7	Superheat, Fahr.	Nil	Nil	200°	100°	200°	100°	260°
8	Vacuum, inches	27	28	27	28	28½	28½	28½
9	Steam per H.P. hour, main engines, lbs.	12·71	11·86	10·4	10·9	9·0	10·1	7·4
10	Steam per hour, main engines, lbs.	274400	258400	225000	238000	211050	220200	167240
11	Steam per hour, all auxiliaries, lbs.	43600	43600	36000	38000	35000	35000	29000
12	Total pounds	318000	302000	261000	276000	246050	255200	196240
13	Equivalent coal per hour, lbs	31800	30200	27200	28200	25700	26000	20650
14	Coal used per H.P. hour, lbs.	1·47	1·385	1·257	1·292	1·205	1218	1·032
15	B.T.U.'s per H.P. hour	20600	19400	17600	18100	16900	17050	14500
16	Distance run in nautical miles per ton of coal	1·355	1·426	1·585	1·53	1·68	1·67	2·09
17	Economy over A, per cent.	...	5	14½	11½	19	18	35
18	Boilers—No.	6 D.E. & 4 S.E.	6 D.E. & 4 S.E.	6 D.E. & 3 S.E.	6 D.E. & 3 S.E.	6 D.E. & 2 S.E.	6 D.E. & 2 S.E.	4 D.E. & 3 S.E.
19	Heating surface, square feet	52000	52000	48750	48750	45500	45500	35750
20	Grate area, square feet	1232	1232	1155	1155	1078	1078	847
21	Draught	Howden's	Howden's	Howden's	Howden's	Howden's	Howden's	Howden's
22	Coal per day, tons	341	324	292	302	276	279	222
23	Eight-days' bunkers, tons	2728	2592	2336	2416	2208	2232	1776
24	Weight of machinery, tons	3675	2965	3610	2900	2445	2605	2310
25	Weight of machinery, plus weight of bunker coal, tons	6403	5557	5946	5316	4653	4837	4086
26	Gain in dead weight, tons	...	846	457	1087	1750	1566	2317
27	Length of machinery space, feet	252	237	234	231	192	201	180
28	Saving in machinery space, feet	...	15	18	21	60	51	[illegible]
[illegible]	Cubic capacity gained, feet	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

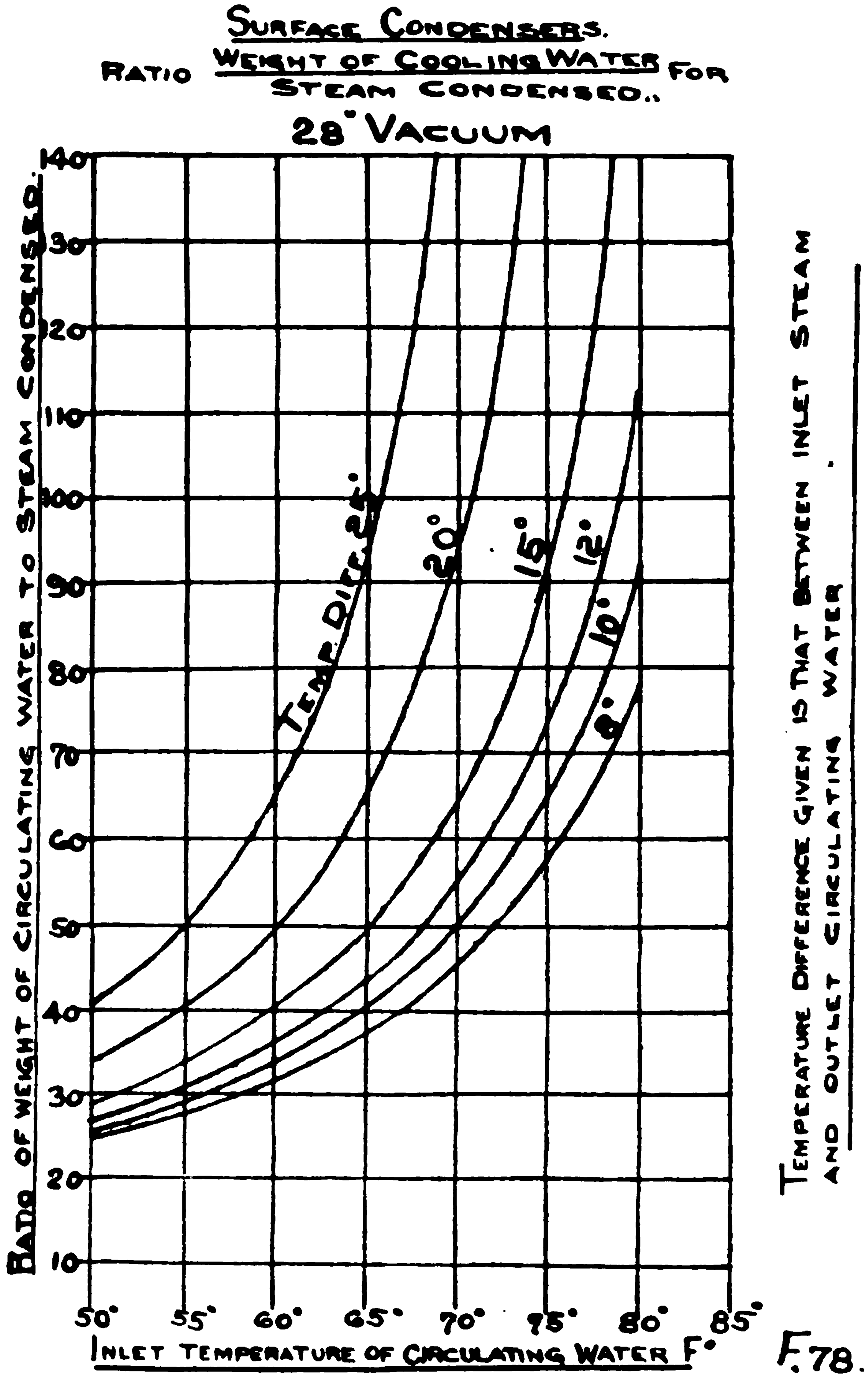
Condensation Water Charts.

The following charts show the ratio of cooling water to steam condensed (*i.e.*, lbs. water required per lb. steam) at various sea temperatures, and temperature differences, and for three conditions of vacuum, 27·5″, 28″, and 28·5″. The charts may be taken to represent modern turbine practice.

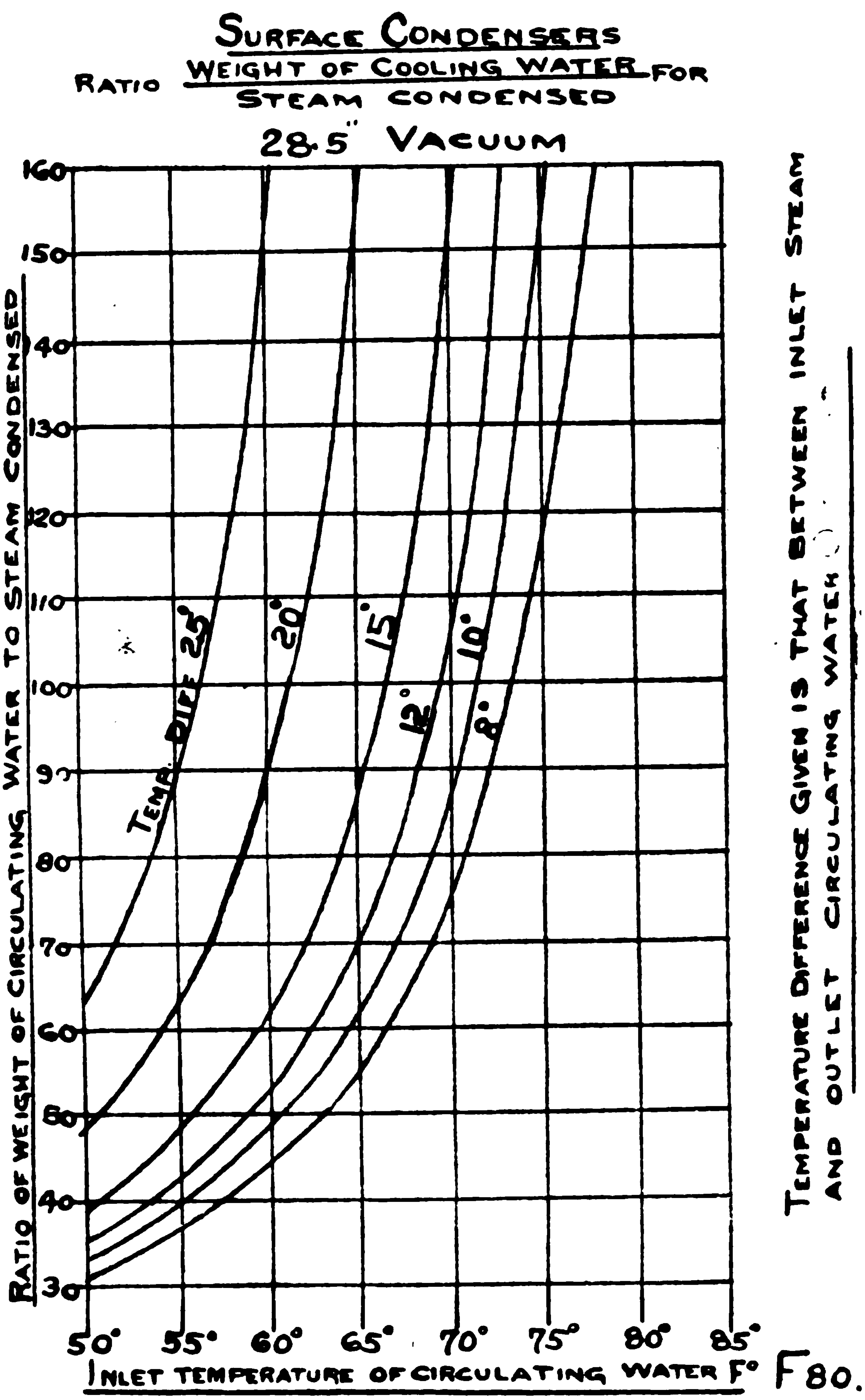


F 77

No. 31.


Surface Condensers.
Ratio Weight of Cooling Water / Steam Condensed for 28" Vacuum
Ratio of weight of circulating water to steam condensed.
140
130
120
110
100
90
80
70
60
50
40
30
20
10
Temp. Diff. 25°
20°
15°
12°
10°
8°
50°
55°
60°
65°
70°
75°
80°
85°
Inlet Temperature of Circulating Water F°
Temperature difference given is that between inlet steam and outlet circulating water.
F. 78.

No. 32.

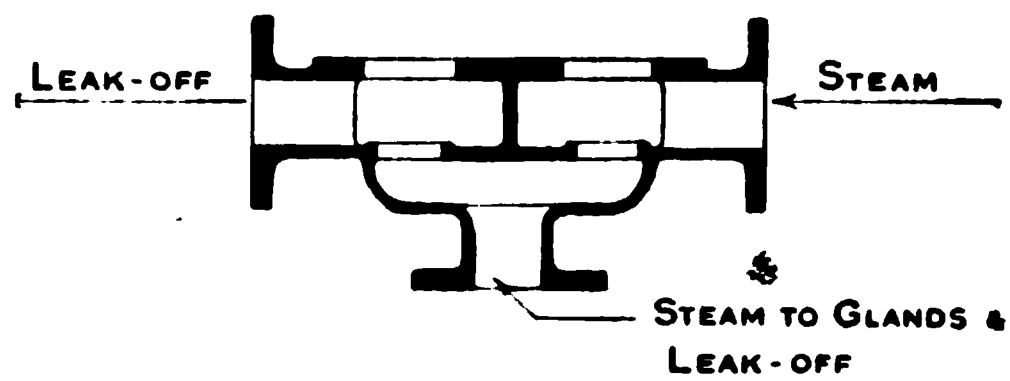


No. 33.

Referring to above chart, suppose the sea water temperature is 60° and the temperature difference between the exhaust steam and discharge is required to be, say, 20°, then the amount of circulating water required per lb. of steam condensed will be 90 lbs., as the plotted line marked 20° cuts the corner of the square marked 90 (on left of chart); and the same holds good for any other sea temperature, and temperature difference. If, however, the temperature difference only requires to be, say, 8°, then the cooling-water per lb. steam will be about 45 lbs., as the plotted line marked 8° cuts the middle of the square between the 40 and 50 lb. level (marked on left of chart).

TRIGONOMETRICAL RATIOS, ETC.

Angle.	Radians.	Sine.	Tangent.	Co-tangent.	Cosine.		
0°	0	0	0	∞	1	1·5708	90°
1	·0175	·0175	·0175	57·2900	·9998	1.5533	89
2	·0349	·0349	·0349	28·6363	·9994	1.5359	88
3	·0524	·0523	·0524	19·0811	·9986	1·5184	87
4	·0698	·0698	·0699	14·3006	·9976	1·5010	86
5	·0873	·0872	·0875	11·4301	·9962	1·4835	85
6	·1047	·1045	·1051	9·5144	·9945	1·4661	84
7	·1222	·1219	·1228	8·1443	·9925	1·4486	83
8	·1396	·1392	·1405	7·1154	·9903	1·4312	82
9	·1571	·1564	·1584	6·3138	·9877	1·4137	81
10	·1745	·1736	·1763	5·6713	·9848	1·3963	80
11	·1920	·1908	·1944	5·1446	·9816	1·3788	79
12	·2094	·2079	·2126	4·7046	·9781	1·3614	78
13	·2269	·2250	·2309	4·3315	·9744	1·3439	77
14	·2443	·2419	·2493	4·0108	·9703	1·3265	76
15	·2618	·2588	·2679	3·7321	·9659	1·3090	75
16	·2793	·2756	·2867	3·4874	·9613	1·2915	74
17	·2967	·2924	·3057	3·2709	·9563	1·2741	73
18	·3142	·3090	·3249	3·0777	·9511	1·2566	72
19	·3316	·3256	·3443	2·9042	·9455	1·2392	71
20	·3491	·3420	·3640	2·7475	·9397	1·2217	70
21	·3665	·3584	·3839	2·6051	·9336	1·2043	69
22	·3840	·3746	·4040	2·4751	·9272	1·1868	68
23	·4014	·3907	·4245	2·3559	·9205	1·1694	67
24	·4189	·4067	·4452	2·2460	·9135	1·1519	66
25	·4363	·4226	·4663	2·1445	·9063	1·1345	65
26	·4538	·4384	·4877	2·0503	·8988	1·1170	64
27	·4712	·4540	·5095	1·9626	·8910	1.0996	63
28	·4887	·4695	·5317	1·8807	·8830	1·0821	62
29	·5061	·4848	·5543	1·8040	·8746	1·0647	61
30	·5236	·5000	·5774	1·7321	·8660	1·0472	60
31	·5411	·5150	·6009	1·6643	·8572	1·0297	59
32	·5585	·5299	·6249	1·6003	·8480	1·0123	58
33	·5760	·5446	·6494	1·5399	·8387	·9948	57
34	·5934	·5592	·6745	1·4826	·8290	·9774	56
35	·6109	·5736	·7002	1·4281	·8192	·9599	55
36	·6283	·5878	·7265	1·3764	·8090	·9425	54
37	·6458	·6018	·7536	1·3270	·7986	·9250	53
38	·6632	·6157	·7813	1·2799	·7880	·9076	52
39	·6807	·6293	·8098	1·2349	·7771	·8901	51
40	·6981	·6428	·8391	1·1918	·7660	·8727	50
41	·7156	·6561	·8693	1·1504	·7547	·8552	49
42	·7330	·6691	·9004	1·1106	·7431	·8378	48
43	·7505	·6820	·9325	1·0724	·7314	·8203	47
44	·7679	·6947	·9657	1·0355	·7193	·8029	46
45	·7854	·7071	1·0000	1·0000	·7071	·7854	45
		Cosine.	Co-tangent.	Tangent.	Sine.	Radians.	Angle.


LEAK-OFF
STEAM
STEAM TO GLANDS &
LEAK-OFF